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# COMPUTER MODELING AND DESIGN OPTIMIZATION OF VERTICAL STEAM CONDENSERS

by

NEIL CHARLES BOURGEOIS
LIEUTENANT, U.S. NAVY

B.E., State University of New York,
Maritime College
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Submitted to the Department of Mechanical Engineering on May 7, 1982 in partial fulfillment of the requirements for the Degrees of Ocean Engineer and Master of Science in Naval Architecture and Marine Engineering

## ABSTRACT

The objective of this study is to investigate Vertical Steam Condenser design optimization, in an effort to develop condensers with minimum weight and volume. Candidate vertical condenser tubes considered were of two types: (1) external axial flutes with smooth internal tubes, and (2) doubly enhanced tubes that include internal (coolant side) heat transfer augmentation. A computer program (VERTCON-2) was developed to be used as a preliminary design tool for selection of the optimal condenser that meets the specified geometrical, and operating parameter limits.

The computer model considers condensation in the presence of non-condensable gases. Also, a method was developed to evaluate thermal resistance of the condenser tube walls due to the axial flutes.

Thesis Supervisor: Warren M. Rohsenow

Title: Professor of Mechanical Engineering

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# TABLE OF CONTENTS


| | | Page |
|---|---|---|
| ABSTRACT | | 2 |
| ACKNOWLEDGEMENTS | | 3 |
| TABLE OF CONTENTS | | 4 |
| LIST OF FIGURES AND TABLES | | 5 |
| CHAPTER I. | INTRODUCTION | 8 |
| CHAPTER II. | HEAT TRANSFER CALCULATIONS | 12 |
| | Nomenclature | 12 |
| II.A | Analysis Summary | 14 |
| II.B | Tube Wall Resistance | 17 |
| II.C | Waterside Heat Transfer Coefficient | 28 |
| II.D | Condensate Heat Transfer Coefficient | 31 |
| II.E | Selection of External Axial Flute Amplitude-to-Pitch (a/p) Ratio | 33 |
| CHAPTER III. | CONDENSATION IN THE PRESENCE OF NON-CONDENSABLE GASES | 37 |
| | Nomenclature | 37 |
| | Heat and Mass Transfer Equations | 39 |
| CHAPTER IV. | PRESSURE DROP CALCULATIONS | 45 |
| IV.A | Tubeside Pressure Drop | 47 |
| IV.B | Shellside Pressure Drop | 49 |
| CHAPTER V. | COMPUTER MODELING | 56 |
| APPENDIX A. | VERTCON-2 COMPUTER PROGRAM DOCUMENTATION | |
| | Data Input | 61 |
| | Data Output | 68 |
| | Program Listing | 72 |
| APPENDIX B. | Reference Data | 106 |
| REFERENCES | | 112 |

## FIGURES


| | | Page |
|---|---|---|
| 1 | VERTICAL ENHANCED TUBE CONDENSER. . . . . | 10 |
| 2 | CROSS SECTION OF VERTICAL FLUTED CONDENSER SURFACE. . . . . | 16 |
| 3 | a. TWO-DIMENSIONAL HEAT FLOW THROUGH AN ELLIPTICAL BOUNDARY . . . . . | 16 |
| | b. HEAT FLOW RESTRICTION MODEL . . . . . | 16 |
| | c. APPLICATION OF HEAT FLOW RESTRICTION MODEL TO AXIAL FLUTED TUBES . . . . . | 16 |
| 4 | CURVILINEAR COORDINATE SYSTEM . . . . . | 19 |
| 5 | STEADY ONE-DIMENSIONAL HEAT TRANSFER THROUGH A THICKWALLED CYLINDRICAL SHELL. . . . . | 19 |
| 6 | COMPARISON OF FLUTE THERMAL RESISTANCE CALCULATIONS . . . . . | 27 |
| 7 | INTERNAL ENHANCEMENT GEOMETRY . . . . . | 29 |
| 8 | $h_c/h_c^*$ VERSUS a/p, CONSTANT FLUTE PITCH CURVES. . | 34 |
| 9 | $h_c/h_c^*$ VERSUS a/p, CONSTANT FLUTE AMPLITUDE AND PITCH CURVES AT $W/W_F$=.1 . . . . . | 34 |
| 10 | $h_c/h_c^*$ AND $R_f/R_f^*$ VERSUS a/p FOR CONSTANT AMPLITUDE (a=.0125in.) AND CONSTANT PITCH (p=.025in.) CURVES. . . . . | 35 |
| 11 | $h_c/h_c^*$ AND $R_f/R_f^*$ VERSUS a/p FOR CONSTANT AMPLITUDE (a=.01in.) AND CONSTANT PITCH (p=.050in.) CURVES. . . . . | 35 |
| 12 | RECOMMENDED AIR EXTRACTION CAPACITY . . . . . | 44 |
| 13 | STEAM DISTRIBUTION PRESSURE ZONES. . . . . | 50 |

FIGURES (CONT'D)


| | | Page |
|---|---|---|
| 14 | FRICTION FACTOR FOR STEAM DISTRIBUTION LOSSES. . . . . | 55 |
| 15 | FRICTION FACTOR FOR STEAM PENETRATION (TUBE BANK) LOSSES. . . . . | 55 |
| 16 | VERTCON-2 OPTIMIZED CONDENSER SIZING PROGRAM FLOWCHART . . . . . | 57 |
| 17 | FRICTION FACTOR CURVES FOR MULTIPLE-HELIX INTERNAL RIDGED TUBES . . . . . | 108 |
| 18 | OPERANDS FOR FRICTION FACTOR EQUATION VERSUS GEOMETRIC ASPECT RATIO e/1 FOR MULTIPLE-HELIX INTERNAL RIDGING | |
| | $p/di \geq 0.36$. . . . . | 109 |
| | $p/di < 0.36$. . . . . | 109 |
| 19 | WATERBOX AND TUBE END LOSSES SINGLE PASS CONDENSERS. . . . . | 110 |
| 20 | WATERBOX AND TUBE END LOSSES DOUBLE PASS CONDENSERS. . . . . | 111 |

# TABLES


| | | Page |
|---|---|---|
| 1 | RECOMMENDED STEAM DESIGN VELOCITIES. . . . . | 51 |
| 2 | SAMPLE INPUT DATA FILE . . . . . | 62 |
| 3 | SAMPLE OUTPUT DATA FILE. . . . . | 68 |
| 4 | PROPERTIES OF MULTIPLE-HELIX INTERNAL RIDGED TUBES. . . . . | 107 |
| 5 | FRICTION FACTOR CHARACTERISTICS OF MULTIPLE-HELIX INTERNAL RIDGED TUBES. . . . . | 108 |

# I. INTRODUCTION

A design optimization procedure for vertical steam condensers is established here to develop condensers with minimum weight and volume. The performance comparison of enhanced steam condensers conducted in Reference-3 suggests the following methodology that provides the basis for this optimization scheme.

The optimization procedure is outlined as follows:

1. Design for the following given condenser operating conditions:
   a. total steam condensed
   b. condenser operating pressure
   c. coolant inlet temperature
   d. coolant flow rate
2. This design is subject to the following specified parameter limits:
   a. maximum condenser length ($L_{max}$)
   b. maximum pumping power ($P_{max}$)
   c. coolant velocity range from a minimum velocity ($V_{min}$) to a maximum velocity ($V_{max}$).
3. For a given heat transfer surface minimum weight and volume will always be achieved for a maximum coolant flow velocity that results in:
   a. condenser length is equal to $L_{max}$ for pumping power less than $P_{max}$, or
   b. pumping power equal to $P_{max}$ for condenser length less than $L_{max}$.
4. Thus, the optimization routine proceeds as follows:
   a. select value for coolant velocity (V)
   b. calculate overall heat transfer coefficient (U) using the equations from Chapter II

c. calculate tube length and the rest of the condenser geometry to finally determine total condenser length ($L_c$)

d. check if pumping power (P) is less than $P_{max}$ and $L_c = L_{max}$

e. if the above conditions for P and $L_c$ are not met, then either change V or enhancement of heat transfer surface

5. a. if the calculated condenser length $L_c < L_{max}$ and $P < P_{max}$ change to less enhanced surface until $L_c = L_{max}$ or $P = P_{max}$

b. if $L_c > L_{max}$ and $P < P_{max}$ increase the surface enhancement until either $L_c = L_{max}$ or $P = P_{max}$

c. if $L_c > L_{max}$ and $P > P_{max}$ then decrease V until either $L_c = L_{max}$ and $P \leq P_{max}$ or $P = P_{max}$ and $L \leq L_{max}$

Further details of this optimization procedure can be found in Chapter V-Computer Modeling.

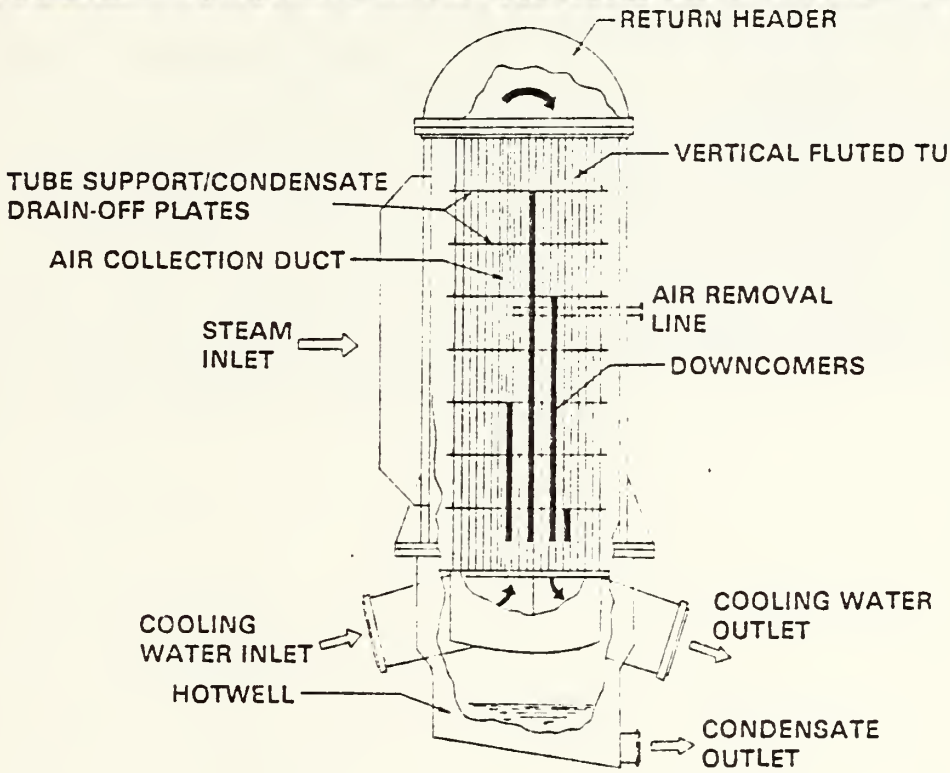
Vertical condenser tubes can provide great flexibility in this design optimization procedure due to the different possible combinations of internal and external enhancement configurations. Thus it is possible to "fine tune" the overall heat transfer coefficient to obtain the proper condenser length.

The procedure that has been outlined here is included in a preliminary design program VERTCON-2 along with the proposed vertical condenser arrangement shown in Figure-1.

Steam enters the condenser shell along a major portion of the tube length. Steam lanes distribute the steam around the tube bundle with radial steam inflow into the tube bundle towards the air cooler section and air removal duct. Steam condenses on the vertical tubes, and the condensate collects in the valley of the tube fluted surface flowing down the tube length. The tube support plates have a dual

FIGURE 1

## VERTICAL ENHANCED TUBE CONDENSER



purpose by also serving as condensate drain-off plates. The condensate is stripped off the tubes and collected on the tube support plates, and then it is removed to the hotwell section via downcomer drainage tubes. Improved condensate control is afforded by this condensate drainage and removal scheme. The tube support/condensate drain-off plates divide the condenser up into sections, where the full depth of tubes in each section see relatively fresh steam. The tubes will see fresh steam through the depth of the tube bundle since the tubes are not subjected to condensate indundation effects because of effective condensate removal and vertical orientation of the tubes. A continuous duct that runs along the total tube length is utilized for air removal. Figure-1 shows a double flow coolant water circuit with the inlet/outlet header located in the bottom of the condenser. The condenser hotwell is located below and also encases the inlet/outlet header. This hotwell location adds to the total height of the condenser.

## II. HEAT TRANSFER CALCULATIONS

Nomenclature

| | |
|---|---|
| A | area ($ft^2$) |
| a | amplitude of the flute (ft) |
| Cp | specific heat (BTU/lbm °F) |
| D,d | diameter (ft) |
| e | helical ridge height (ft) |
| f | friction factor $\left[=(\Delta P/\rho)(D/L)(2g_c/V^2)\right]$ |
| G | mass flux ($lbm/ft^2hr$) |
| $g_c$ | gravitational constant ($lbm\ ft/lbf\ hr^2$) |
| h | heat transfer coefficient ($BTU/hr\ ft^2\ °F$) |
| $h_{fg}$ | latent heat of vaporization (BTU/lbm) |
| K | thermal conductivity ($BTU/hr\ ft^2\ °F$) |
| L | length (ft) |
| l | lead of ridge (axial distance per 360° turn)(ft) |
| m | operand for friction factor equation |
| Nu | Nusselt number |
| P | pressure ($lbf/ft^2$) or (in-hg-abs) |
| p | pitch of ridging or flute (ft) |
| Pr | Prandtl number |
| Q | heat flow (BTU/hr) |
| r | operand for friction factor equation |
| Re | Reynolds number |
| $Re_f$ | flooding Reynolds number for flute $\left[=4W_f/\mu X_L\right]$** |
| $R_f$ | thermal resistance of axial flute ($hrft^2\ °F/BTU$) |
| $R_s$ | thermal resistance of scale ($hrft^2\ °F/BTU$) |
| $R_w$ | thermal resistance of tube wall ($hrft^2\ °F/BTU$) |
| St | Stanton number |
| T | temperature (°F) |
| $\Delta T$ | $T_{sat}-T_w$ (°F) |
| $\Delta T_{lm}$ | mean overall temperature difference (°F) |
| U | overall heat transfer coefficient ($BTU/hrft^2\ °F$) |

$U^+$ Dimensionless velocity $\left[=U/U^*\right]$ *

$U^*$ friction velocity $\left[=\sqrt{\tau_o g_c/\rho}\right]$ *

V velocity (ft/hr) or (ft/sec)

$W_f$ flooding axial mass flow of condensate per flute (lbm/hr)

$X_L$ half-perimeter length of flute (ft)

Subscripts

b fluid at the bulk temperature ($^\circ$F)

c condensate

f fluid or flooding

i inside, or inlet

n nominal

o outside, or outlet

s increment, or section

sat saturation

w wall

Superscripts

$^+$ dimensionless parameter

Greek Symbols

$\alpha$ height of the condensate in the center of the flute (ft)

$\gamma$ fragment of $U_e^+ \left[= -2.5\ln(2e/d_i) + 3.75\right]$ *

$\lambda$ dimensionless group $= \frac{4\rho^2}{\mu^2} g_c \frac{(\alpha_o)^4}{X_L}$ **

$\mu$ dynamic viscosity (lbm/hr ft)

$\rho$ density ($lbm/ft^3$)

$\sigma$ surface tension (lbf/ft)

$\Omega$ non-dimensional group $^+$

$\tau$ apparent wall shear stress

* Reference (6)

** Reference (4)

$^+$ Reference (1)

## II.A ANALYSIS SUMMARY

The rate of heat flow Q over an entire heat exchanger is related to the mean overall temperature difference $\Delta T_{lm}$ and the total heat transfer area A by the overall heat transfer coefficient U.

$$Q = UA\ \Delta T_{lm} \qquad \text{(II-1)}$$

From the analysis of externally ridged tubes it proves convenient to base the heat transfer coefficient U on the surface area of a smooth tube having an outside diameter $D_n$ equal to the diameter measured over the mid-height of the external flutes (see Figure-2 for external flute geometry).

$$U_n = \frac{Q}{\pi D_n L}\ \Delta T_{lm} \qquad \text{(II-2a.)}$$

where: $$D_n = D_w + 2a \qquad \text{(II-2b.)}$$

The overall U depends on the resistances in series; between cooling water and the tube wall, within the tube wall, and between the tube wall and the working fluid. Fouling resistance on either side of the tube wall can be combined in one term $R_s$.

Tube wall resistance is considered to be the summation of an equivalent smooth tube resistance and flute resistance. Tube wall resistance for a smooth (non-enhanced) tube can be expressed as:

$$R_w = \frac{1}{h_w} = \frac{\ln(D_o/D_i)D_{ref}}{K \quad 2} \qquad \text{(II-3)}$$

Where $D_{ref}$ is the reference diameter on which the overall U is based on. For a fluted tube the overall U is based

on the nominal diameter $D_n$. As will be discussed in the next section , the flute resistance can be written:

$$R_{flute} = \frac{\ln\left[7.985(a)\cdot 7688\left(\frac{a}{p}\right)^{-.0242} + 1\right] D_{ref}}{2k} \qquad \text{(II-4)}$$

Therefore, the total tube wall resistance for the fluted tube can be written:

$$R_w = \frac{\ln\left[D_w/D_i\left(7.985(a)\cdot 7688\left(\frac{a}{p}\right)^{-.0242} + 1\right)\right] D_n}{2k} \qquad \text{(II-5)}$$

Therefore, the concept of resistances in series yields:

$$\frac{1}{U_n} = \frac{D_n/D_i}{h_f} + R_s + R_w + \frac{1}{h_c} \qquad \text{(II-6)}$$

The overall heat transfer coefficient $U_n$ in Equation (II-6) does not consider non-condensable gas effects. Condensing in the presence of non-condensable gases will be taken up in Chapter V.

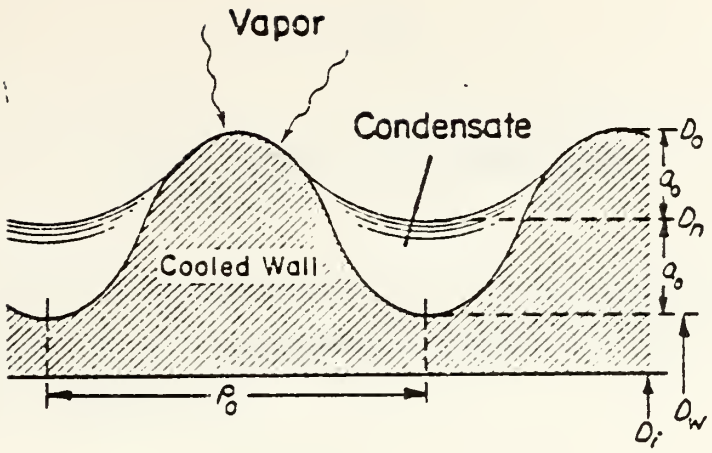



FIGURE 2
Cross Section of Vertical Fluted Condenser Surface

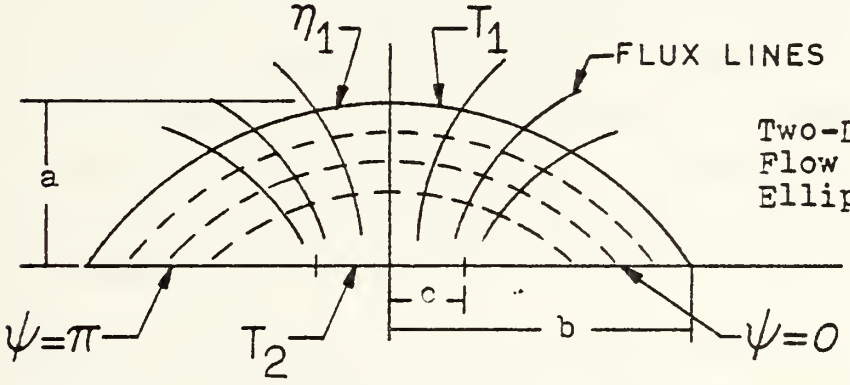



FIGURE 3a.
Two-Dimensional Heat Flow Through an Elliptical Boundary

$\eta_i$ -Constant Temperature Line
$\eta,\psi,Z$ -Coordinates
Z -Coordinate into Page

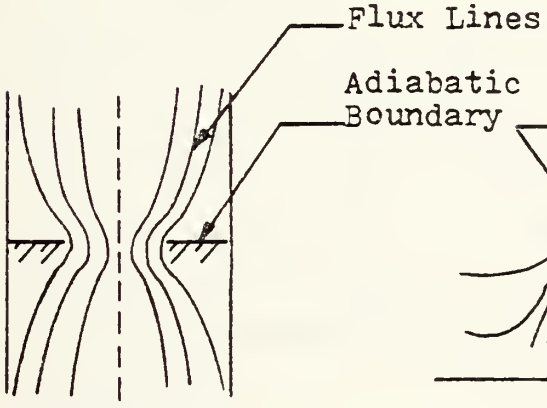



FIGURE 3b.
Heat Flow Restriction Model

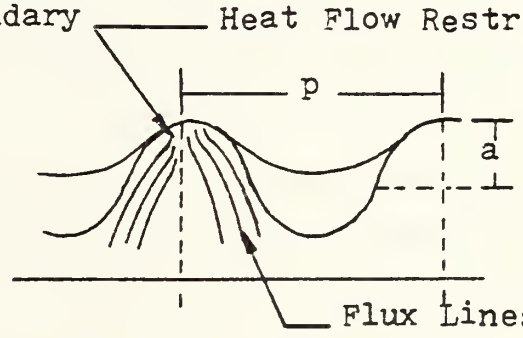



FIGURE 3c.
Application of Heat Flow Restriction Model to Axial Fluted Tubes

## II.B TUBE WALL RESISTANCE

One method to evaluate tube wall resistance for fluted tubes would be to use Equation-3, where $D_o$ and $D_{ref}$ equal the nominal diameter $D_n$.

$$R_w = \frac{\ln(D_n/D_i)D_n}{2K} \qquad \text{(II-7)}$$

This formulation is not sensitive to flute geometry such as changes in amplitude-to-pitch (a/p)ratio. In comparing Equation-7 with finite element formulation differences of up to 30 percent in flute resistance are observed. This could amount to 10 percent difference when considering total tube wall resistance for tubes made with low conductivity materials such as titanium with thin tube wall dimensions.

What is proposed here is to model flute and condensate layer as a heat flow restriction, by describing two-dimensional heat flow through an elliptical boundary. (Figure-3a) This serves as an accurate model for a heat flow restriction as shown in Figure-3b.

The procedure is to apply this same model to describe the heat flow through the axial flutes (Figure-3c).

Some aspects of curvilinear coordinate systems (7) will be reviewed in order to develop nomenclature to be used in later equations.

Rectangular coordinates x,y,z, can be expressed in terms of new coordinates $u_1$, $u_2$, $u_3$ by the equations: (see Figure-4)

$$x = x(u_1, u_2, u_3)$$

$$y = y(u_1, u_2, u_3)$$

$$z = z(u_1, u_2, u_3)$$

The coordinates are chosen such that $u_1$, $u_2$, $u_3$, are orthogonal curvilinear coordinates. From examination of Figure-4 it can be seen that if a particle moves from point P in such a way that $u_2$ and $u_3$ are held constant and only $u_1$ varies, then a curve in space is generated.

Let $\bar{r}$ represent the position vector of a point P in space:

$$\bar{r} = x\hat{i} + y\hat{j} + z\hat{k}$$

where $\hat{i}$, $\hat{j}$, and $\hat{k}$ are unit vectors in the x, y and z directions respectively.

Then a tangent vector to $u_1$ curve at point P is given by:

$$\bar{U}_1 = \frac{\partial \bar{r}}{\partial u_1} = \frac{\partial \bar{r}}{\partial s_1} \frac{dS_1}{du_1} \qquad \text{(II-8)}$$

$S_1$ = arc length along the $u_1$ curve

$\frac{\partial \bar{r}}{\partial s_1}$ is a unit vector

thus: $\bar{U}_1 = h_1 \bar{u}_1$

where $\bar{u}_1$ is the unit vector tangent to $u_1$ in the direction of increasing arc length,

thus: $h_1 = \frac{dS_1}{du_1}$ which is the length of $\bar{U}_1$

Similarly for the other coordinates:

$$\bar{U}_1 = h_1\bar{u}_1 \quad \bar{U}_2 = h_2\bar{u}_2 \quad \bar{U}_3 = h_3\bar{u}_3$$

$$h_1 = \frac{dS_i}{du_i} \quad i = 1,2,3 \quad dS_i = h_i du_i \quad i = 1,2,3 \qquad \text{(II-9)}$$

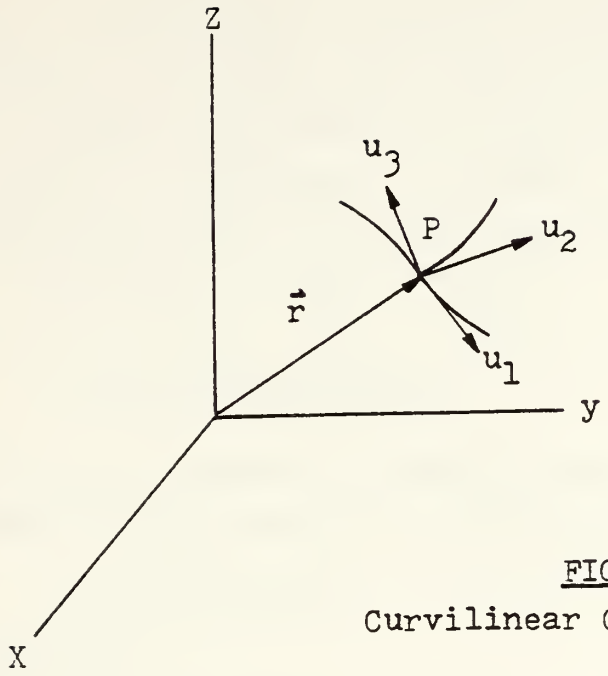



FIGURE 4
Curvilinear Coordinate System

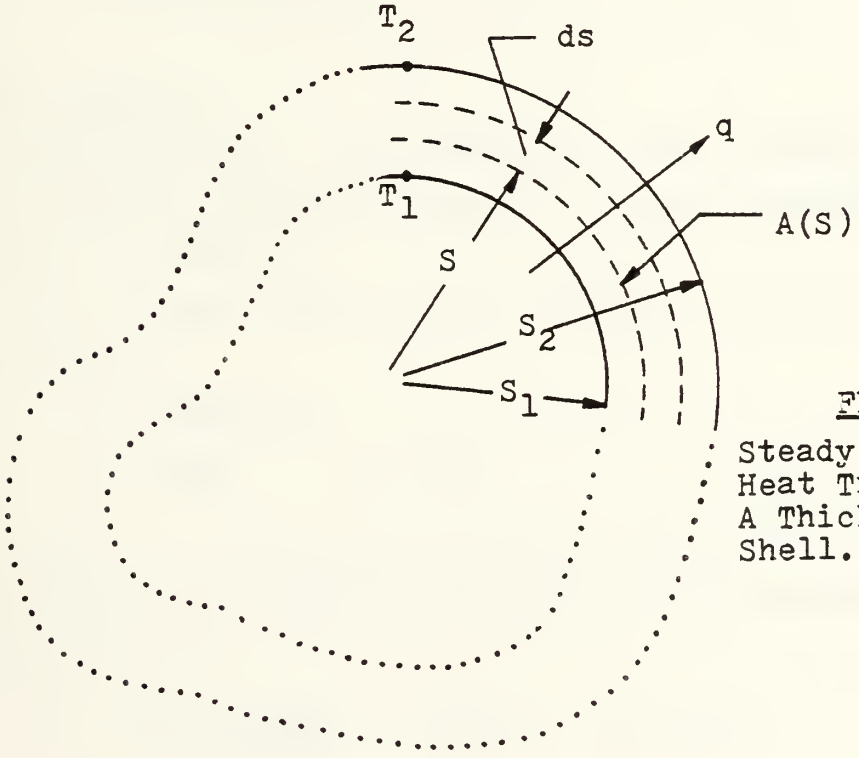



FIGURE 5
Steady One-Dimensional Heat Transfer Through A Thickwalled Cylindrical Shell.

The unit vectors $\bar{u}_1$, $\bar{u}_2$ and $\bar{u}_3$ form a right-handed orthogonal curvillinear coordinate system. The vectors $\bar{U}_1 du_1$, $\bar{U}_2 du_2$, and $\bar{U}_3 du_3$ are mutually perpendicular vectors having as their length the arc-length differentials $dS_1$, $dS_2$, and $dS_3$. Therefore, an element of volume can be determined from the following vector product:

$$dV = \bar{U}_1 du_1 \times \bar{U}_2 du_2 \cdot \bar{U}_3 du_3$$

$$dV = h_1 h_2 h_3 du_1 du_2 du_3 \qquad \text{(II-10)}$$

Also, a vector element of surface area dA, on the surface $u_1$ = constant is given by the vector product:

$$dA_2 = h_2 h_3 du_2 du_3 \qquad \text{(II-11a)}$$

and similarly:

$$dA_2 = h_3 h_1 du_3 du_1 \qquad \text{(b)}$$

$$dA_3 = h_1 h_2 du_1 du_2 \qquad \text{(c)}$$

GENERAL PROBLEM

Consider steady one-dimensional heat transfer through a thick-walled cylindrical shell a section of which is displayed in Figure-5.

First assume rate of heat transfer is constant at every section.

$$Q = \text{constant}$$

$$Q = -KA\frac{dT}{ds} \qquad \text{(II-12)}$$

Where S is defined in Figure-5, integrating Equation (II-12) between inner and outer surfaces with the assumption K is constant yields:

$$Q = \frac{T_1 - T_2}{(1/K)\int_{s_1}^{s_2} ds/A(s)} = \frac{\Delta T}{R} \qquad \text{(II-13)}$$

where R is heat transfer resistance.

thus: $$R = \frac{\Delta T}{Q} = \frac{1}{K}\int \frac{ds}{dA(s)} \qquad \text{(II-14)}$$

Equation (II-14) is only applicable when heat flux is constant across the heat transfer area. In many geometries the distribution of heat flux across the heat transfer area may not be uniform, but equation (II-14) can be expanded for variable heat flux (16).

Equation (II-14) can be rewritten:

$$Q = -K\iint_A \frac{dT}{ds_1}\, ds_2 ds_3 \qquad \text{(II-15)}$$

where: $ds_1 = h_1 du_1$

$ds_2 = h_2 du_2$

$ds_3 = h_3 du_3$

thus: $$\frac{dT}{ds_1} = \frac{1}{h_1}\,\frac{dT}{du_1}$$

and substituting into Equation (II-15):

$$Q = -K\,\frac{dT}{du_1}\iint_A \frac{h_2 h_3}{h_1}\, du_2 du_3$$

thus: $$R = \frac{\Delta T}{Q} = \frac{1}{K}\int_{(u_1)_I}^{(u_1)_{II}} \frac{du_1}{\iint \frac{h_2 h_3}{h_1}\, du_2 du_3} \qquad \text{(II-16)}$$

Equation (II-16) is applicable to variable heat flux distribution over the area of heat transfer. But this equation is restricted to one-dimensional, steady state analysis, where temperature is a function of $u_1$ only.

Equation (II-16) can be applied to the model of two dimensional heat flow through an elliptical boundary.

Referring to Figure-3 and Equation (II-16) the following expressions are obtained:

$\eta_i$ = constant temperature lines

$\eta, \psi, z$ coordinates, where z coordinate is into page

$$h_i = \left| \frac{\partial \bar{r}}{\partial u_i} \right| \quad \text{where} \quad \bar{r} = x\hat{i} + y\hat{j} + z\hat{k}$$

thus:
$$h_i^2 = \left(\frac{\partial x}{\partial u_i}\right)^2 + \left(\frac{\partial y}{\partial u_i}\right)^2 + \left(\frac{\partial z}{\partial u_i}\right)^2 \qquad \text{(II-17)}$$

in this case:

$$x = c \cosh\eta \cos\psi$$
$$y = c \sinh\eta \sin\psi$$
$$z = z$$

from Equation (II-17):

$$h_1 = h_2$$
$$h_3 = 1$$

substituting into Equation (II-16)

$$R = \frac{1}{K} \int_0^{\eta_1} \frac{d\eta}{\int_0^{\pi} \int_0^{L} d\psi dz}$$

and:
$$R = \frac{\eta_1}{K\pi L} \qquad \text{(II-18)}$$

$x = c \cosh\eta \cos\psi$

at $\psi = 0$

$x = b = c \cosh\eta$

and similarly:

$y = c \sinh\eta \sin\psi$

at $\psi = \pi/2$

$y = a = c \sinh\eta$

therefore:

$$\eta_1 = \sinh^{-1}\left(\frac{a}{c}\right)$$

and from a fundamental identity:

$$\eta_1 = \ln\left[\frac{a}{c} + \sqrt{\left(\frac{a}{c}\right)^2 + 1}\right] \qquad \text{(II-19)}$$

For axial fluted tubes Equations (II-18) and (II-19) will take the form:

$$R = \frac{\eta}{K\pi L} \qquad \text{(II-20a.)}$$

$$\eta = \ln\left[C\left(\frac{a}{p}\right) + \sqrt{C^2\left(\frac{a}{p}\right)^2 + 1}\right] \quad \text{(b)}$$

where C is defined as the flute resistance coefficient.

Equation (II-20) is in good agreement with actual physical situation:

If $a=o$ this implies $\eta = \ln(1) = o$, thus $R=o$

1. Therefore a flute of zero amplitude has zero thermal resistance.

2. As flute amplitude increases the flute resistance also increases.

Equation (II-20) can be simplified as follows:

1. Amplitude-to-pitch (a/p) ratios for fluted tubes are in the range of .1-to-.5.

2. The flute resistance coefficient (C) will have an approximate value of $C = .5$ thus,

$$\sqrt{C^2\left(\frac{a}{p}\right)^2 + 1} \approx 1$$

and equations (II-20) simplifies to:

$$R = \frac{\ln\left[C\left(\frac{a}{p}\right)+1\right]}{K\pi L} \quad \text{(II-21)}$$

To express thermal resistance in units of ($hrft^2$ F/BTU) multiply R from Equation (II-21) by a reference area ($A_{ref}$). This reference area should be the same reference area that the overall heat transfer coefficient is based on. thus:

$$R = \frac{A_{ref}(\Delta T)}{Q} \quad (hrft^2\,{}^\circ F/BTU) \quad \text{(II-22)}$$

$$R_f = \frac{\ln\left[C\left(\frac{a}{p}\right)+ 1\right]D_{ref}}{2K} \quad \text{(II-23)}$$

Values for flute resistance are calculated from a finite element formulation using the ADINAT (2) finite element program. The flute resistances obtained from ADINAT are substituted into Equation (II-23), and the values for flute resistance coefficient (C) are determined at different values of a/p, and for various tube dimensions. One result is that C is a function of both flute amplitude a, and a/p ratio. Thus C as a function of a and a/p was correlated using multiple non-linear regression (15) analysis.

Then a final formulation for flute thermal resistance was established in the following form:

$$R_f = \frac{\ln\left[b_o(a)^{b_1}\left(\frac{a}{p}\right)^{(1+b_2)} + 1\right]D_{ref}}{2K} \quad \text{(II-24)}$$

where $C = b_o(a)^{b_1}\left(\frac{a}{p}\right)^{b_2}$

$b_o$, $b_1$, $b_2$ are correlation coefficients resulting from the multiple non-linear regression.

The convection heat transfer coefficient along the perimeter of the flute is required for input into ADINAT. The distribution of convection heat transfer coefficient was estimated by first calculating the condensate film thickness ($\delta$) along the perimeter using a procedure established by BARNES (4). The convection heat transfer at a particular point along the flute perimeter can be approximated by:

$$h_c = \frac{K}{\delta} \quad (BTU/hrft^2\,^\circ F)$$

where K is the thermal conductivity of the condensate, and $\delta$ is the condensate film thickness.

As stated before, the finite element results, along with regression analysis were used to determine the value of C shown below:

$$C = b_o(a)^{b_1}\left(\frac{a}{p}\right)^{b_2} \qquad \text{(II-25)}$$

$b_o = 7.985$

$b_1 = .7688$

$b_2 = -1.0242$

Finally, equation (II-24) can be written:

$$R_f = \frac{\ln\left[7.985(a)^{.7688}\left(\frac{a}{p}\right)^{-.0242}+1\right]D_{ref}}{2K} \qquad \text{(II-26)}$$

The final results of this analysis is depicted in Figure ( 6). Curve-1 represents flute thermal resistance as a function of a/p ratio for constant amplitude flutes. In Curve-2 both flute amplitude and a/p ratio are varying, but flute pitch is held constant.

Also, compared on this plot are the flute thermal resistance calculated from the finite element results, from Equation (II-7) based on nominal tube diameter, from Equation (II-26) based on heat flow restriction model. For ease of comparison thermal resistance have been normalized on the maximum range of thermal resistance considered, which is denoted $R_f^*$. Figure-16 shows that the discrepancy in value between the finite element solution and result based on nominal tube diameter from Equation (II-7) increases with increasing flute amplitude and a/p ratio. Both Curve-1 and Curve-2 together cover a broad range of flute amplitude and a/p values to demonstrate the fit of Equation (II-26).

From the results that were depicted in Figure-6 indicate that Equation (II-26) accurately models the flute thermal resistance. This equation is based on finite element solution which must also be checked to ascertain whether it is a valid model. If the finite element model is developed methodically from a simple configuration where the results are known, to the more complicated model, then confidence can be placed on the results. In this case accurate specification of convection heat transfer coefficients was important to the accuracy of the finite element solution. Also, a simple check can be made to determine if the finite element grid has the proper number of elements to model the problem. This is done by checking the heat flux approaching a boundary node from different elements. If the grid is sufficient there should be no jump in the value of total heat flux across the boundary at a specified location. These general guidelines were followed in modeling tube wall resistance for axially fluted tubes.

Curve-1 - Constant a (a= .002 ft.)
Curve-2 - Constant p (p= .00625 ft.)
——————— Finite Element Solution
- - - - - - - - - Equation (II-26) Heat Flow Restriction Model
—— — —— Equation (II-7) Average Flute Resistance
$R^* = 4.03 \times 10^{-4}$ ($hr ft^2 °F/BTU$)

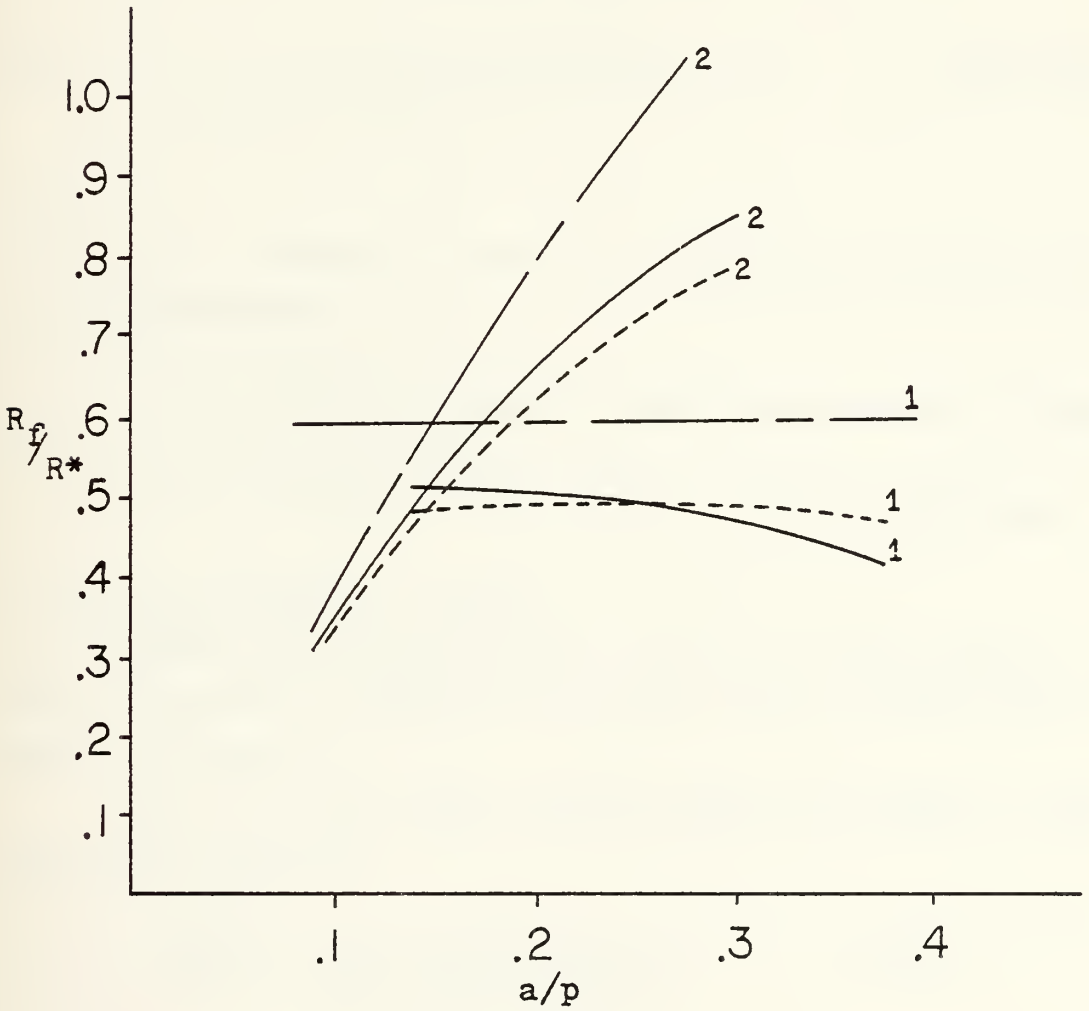



FIGURE 6

Comparison of Flute Thermal Resistance Calculations

## II.C WATERSIDE HEAT TRANSFER COEFFICIENTS

### 1. Smooth Internal Tube

For the smooth internal tube the McAdams correlation was used for determination of the heat transfer coefficient for cooling water.

$$\left[\frac{hD}{K_b}\right] = 0.023\left[\frac{GD}{\mu_b}\right]^{0.8}\left[\frac{\mu Cp}{k}\right]^{0.4} \qquad \text{(II-27a.)}$$

or:

$$h_f = 0.023\,\frac{K_b}{D}\left[Re_D\right]^{0.8}\left[Pr\right]_b^{0.4} \qquad \text{(II-27b.)}$$

where:

(a) all fluid properties are evaluated at the bulk fluid temperature;

(b) $2300\ Re_D\ 10^7$ where;

$Re_D$= Reynolds number based upon hydraulic diameter

(c) $0.5\ Pr_b\ 120$ where;

$Pr_b$= Prandtl number based on bulk temperature

### 2. Helical Internal Ridging

Internal tube augmentation was also investigated, with the application of integral multiple helix ridging (13) as shown in Figure-7.

Friction factor data was correlated by means of the following equation:

$$\sqrt{\frac{f}{8}} = \frac{1}{2.46\ \ln\left[r+(7/Re)^m\right]} \qquad \text{(II-28)}$$

For the roughness of the helical internal ridge both r and m are treated as variables, and are tied in with tube geometry.

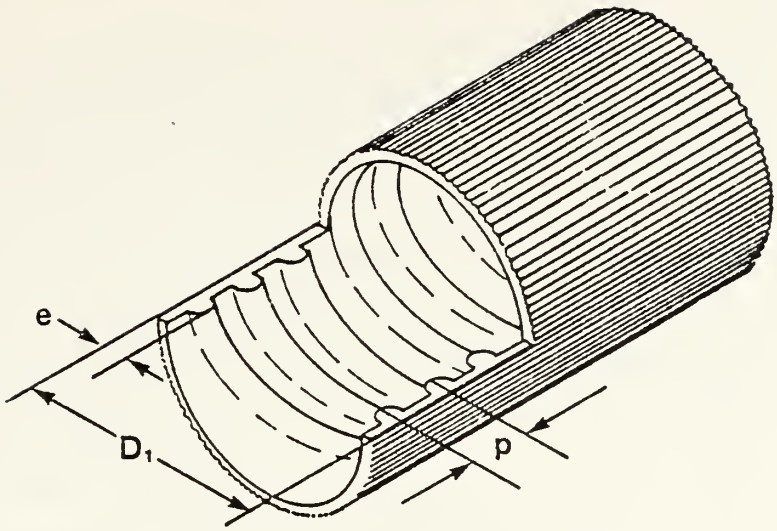
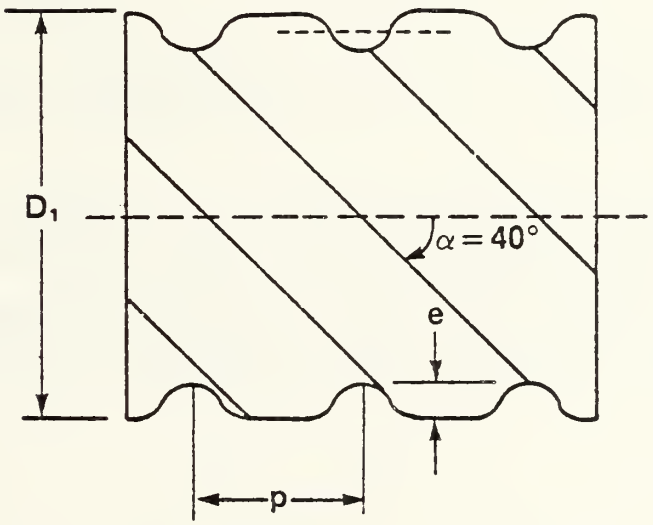



FIGURE 7
Internal Enhancement Geometry



Both m and r vary with the dimensionless parameter e/l, where e is the ridge height and l is the lead of the ridge. In reference (13) WITHERS has made a distinction based on the criterion p/d = 0.36 in correlating the friction behavior to the internal geometry. It has been proposed that a shift in flow behavior occurs at p/d = 0.36. For higher values of p/d a greater degree of swirling could occur, compared with cascading of flow that takes place if p/d is less than 0.36.

The heat transfer correlation equation developed from data for tubes of various configurations, and solved for the Stanton number becomes:

$$St = \frac{\sqrt{f/8}}{5.68(e/p)^{-1/8}\sqrt{Pr}\left[(e/d_i)Re\sqrt{f/8}\right]^{0.136} + \gamma} \qquad \text{(II-29a.)}$$

where: $\gamma = -\left[2.5 \ln(2e/d_i) + 3.75\right]$ (II-29b.)

Then, $h_f$ becomes:

$$h_f = \frac{K_c Re\ Pr\ St}{d_i} \qquad \text{(II-30)}$$

These equations are applicable to Reynolds number range 10,000-120,000 and Prandtl number range 4-10.

See Appendix-B for tube data, and friction factor characteristics of multiple-helix internal ridged tubes.

## II.D CONDENSATE HEAT TRANSFER COEFFICIENT

In reference-2 BARNES developed an equation for the average value of condensate heat transfer coefficient ($h_c$) for an externally fluted tube of length L. In this formulation $h_c$ depends upon physical properties of the condensing fluid geometric factors, and mass flow rate of condensate in the flutes.

$$\bar{h}_c = .6027\left[\frac{h_{fg}W_f}{L\Delta T}\right]^{.0074} \frac{a^{.2307}(Nu_o\Omega^{\frac{1}{4}})}{p} \cdot 9226\left[\frac{K^3\rho\sigma h_{fg}g_c}{\mu\Delta T}\right]^{.2307} \qquad (II\text{-}31)$$

The non-dimensional group $Nu_o\Omega^{\frac{1}{4}}$ defined in Reference-2 is a function of flute amplitude-to-pitch (a/p) ratio. It should be noted in Equation-31 the parameter L is the length of the tube between condensate drain off plates.

The parameter $W_f$ is the flooding axial mass flow of condensate per flute (LBM/hr) which is also a function of physical properties of the fluid, and tube geometry. In Reference-7, PANCHEL and BELL defined the following flooding Reynolds number for condensate flow in the axial flutes:

$$Re_f = \frac{4Wf}{\mu X_L} \qquad (II\text{-}32)$$

and an additional non-dimensional group:

$$\lambda = \frac{4\rho^2}{\mu^2} g_c \frac{(\alpha_o)^4}{X_L} \qquad (II\text{-}33)$$

where $\alpha_o = 2a$ in the case where the flute is flooded. Also, the following correlation was developed for $\lambda_f$ based on the

half-perimeter length ($X_L$) of the flute:

$$\lambda_f = 36(a/p)\exp(3.33\ a/p)\ Re_f \qquad (II-34)$$

Substituting equations (32)and 33)into equation (34) and solving for $W_f$ yields:

$$W_f = \frac{8}{9} g_c \frac{\rho^2}{\mu} a^3 p \exp\left[-3.33\left(\frac{a}{p}\right)\right] \qquad (II-35)$$

where $W_f$ is now based on full perimeter length of the flute.

The non-dimensional group $Nu_o \Omega^{\frac{1}{4}}$ can also be defined by the following correlation:

$$Nu_o \Omega^{\frac{1}{4}} = b_o + \sum_{k=1}^{4} b_k (a/p)^k \qquad (II-36)$$

where $b_o$, $b_k$ are the correlation coefficients:

$b_o = 3.661218$

$b_1 = 6.4526$

$b_2 = -15.265$

$b_3 = 16.14543$

$b_4 = -6.561166$

Equation (II-36) was obtained from multiple linear regression, using techniques of Reference-15.

## II.E SELECTION OF EXTERNAL ENHANCEMENT GEOMETRY

The heat transfer performance of various vertical tubes with different levels of external enhancement were examined. This was done to provide guidance in the selection of external enhancement geometry. The basic parameters that need to be specified are:

1. Flute amplitude (a)
2. Flute amplitude-to-pitch (a/p) ratio, and thus flute pitch (p) is also set.
3. The ratio of condensation rate-to-flooding condensation rate ($W/W_f$).

The following plots of $h_c/h_c^*$ versus a/p ratio (Figure-8) illustrate the relationship between these basic parameters. The variable $h_c$ is the condensing heat transfer coefficient, which is normalized in the plots by the maximum value of $h_c$ (denoted $h_c^*$). The curves of Figure-8 denote lines of constant flute pitch (p) where curve-1 is for $W/W_F$= 0.1 and curve-2 is for $W/W_F$= 0.25. It can be observed that greater enhancement is achieved with larger values of a/p ratios, and smaller values of flute pitch. In Figure-9 a line of constant flute amplitude is also plotted at $W/W_f$=.1. Figure-9 shows that increased enhancement is more readily achieved from increasing a/p ratio by reducing flute pitch.

Increased steamside enhancement cannot be achieved without paying a cost. The cost in this case is increased tube wall resistance. In Figure-10 normalized flute resistance for constant amplitude flutes (curve-1, a= .0125") and constant pitch flutes (curve-2, p= .025") are plotted along with heat transfer data for $W/W_F$= .1. Figure-11 is a similar plot except curve-1 is for a= .01, and curve-2 is for p= .050. Flute thermal resistance was calculated using

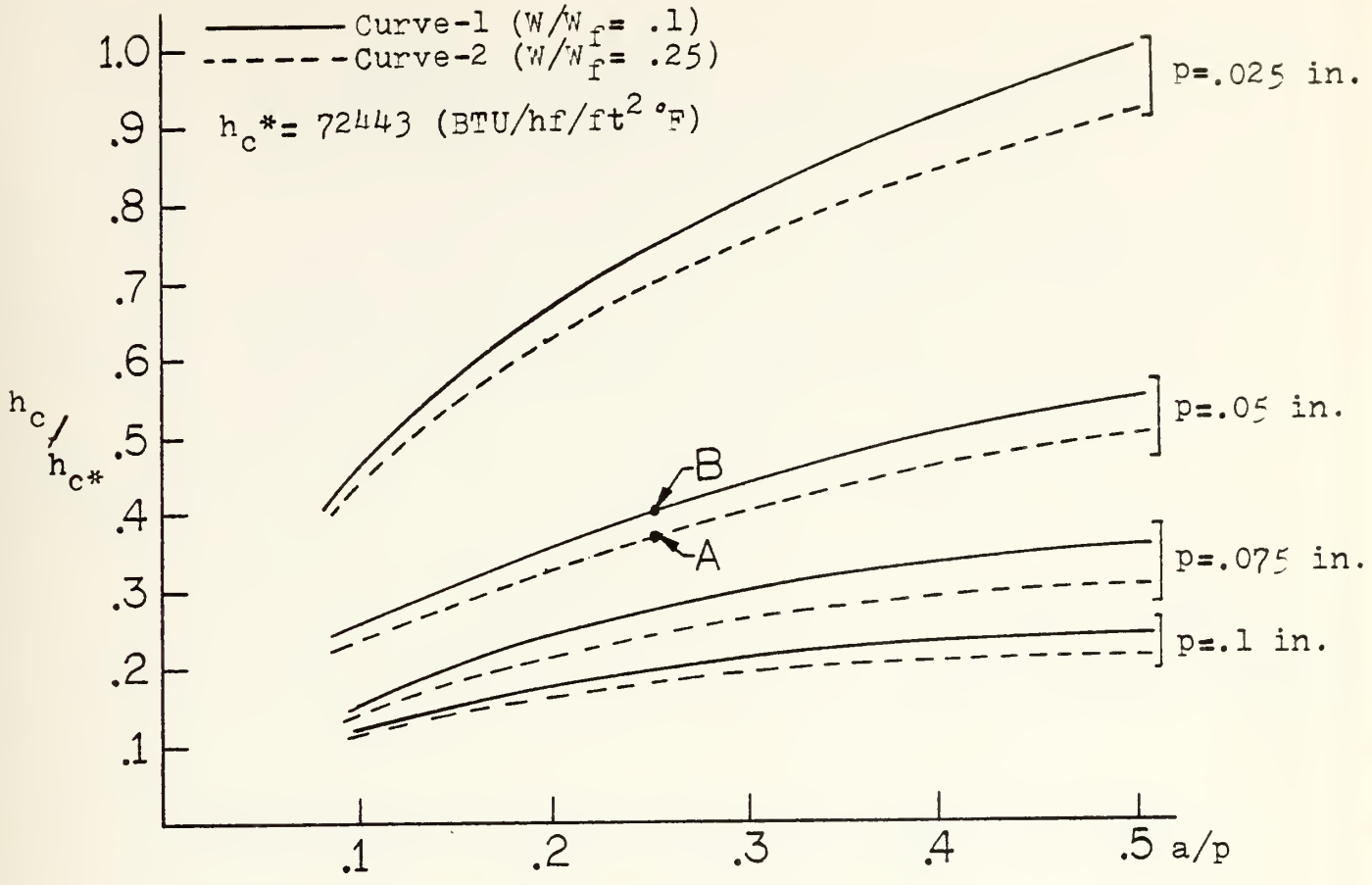



FIGURE 8 - Constant Flute Pitch Curves

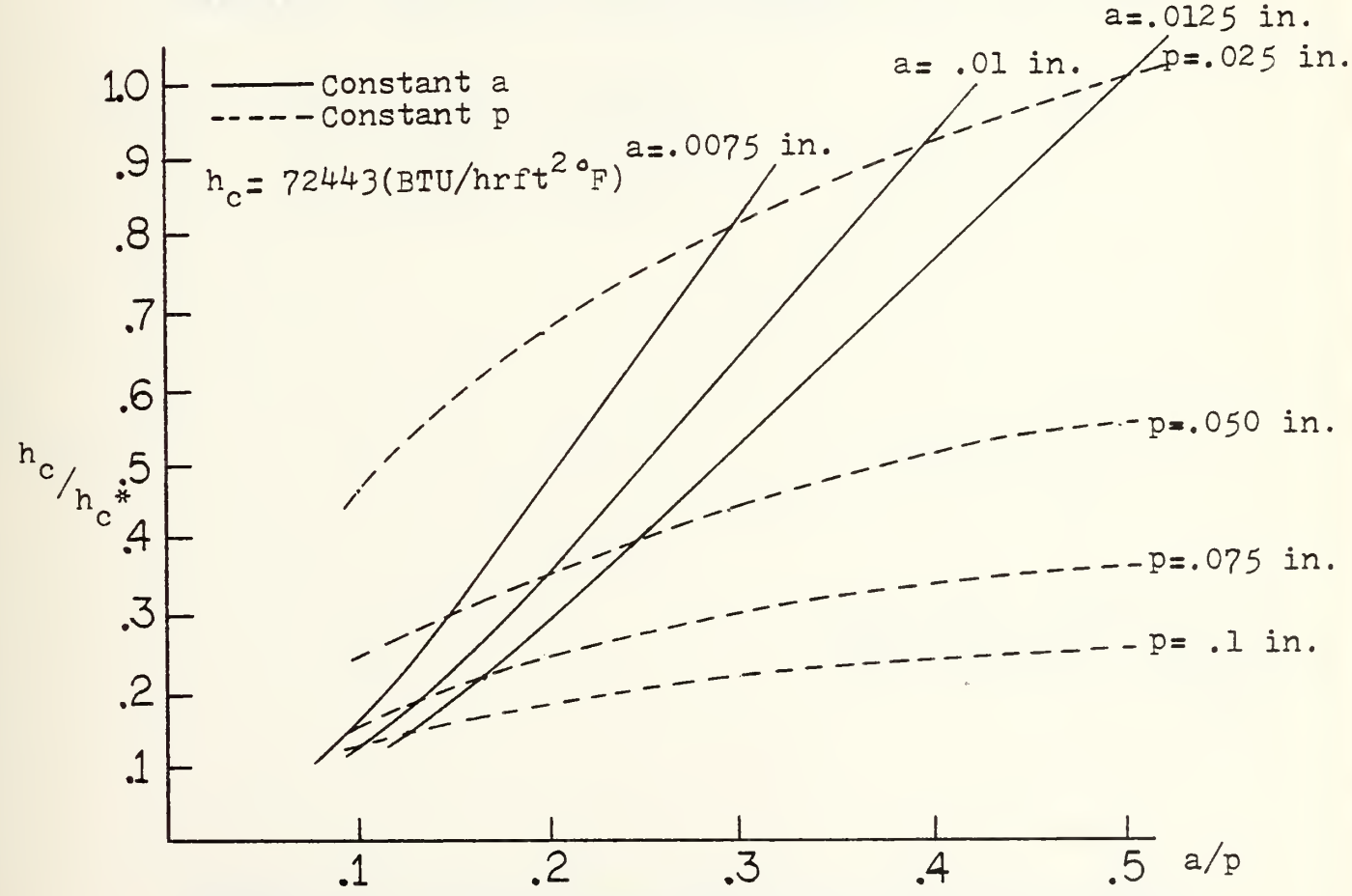



FIGURE 9 - Constant Flute Amplitude and Pitch Curves at $W/W_F$ = .1

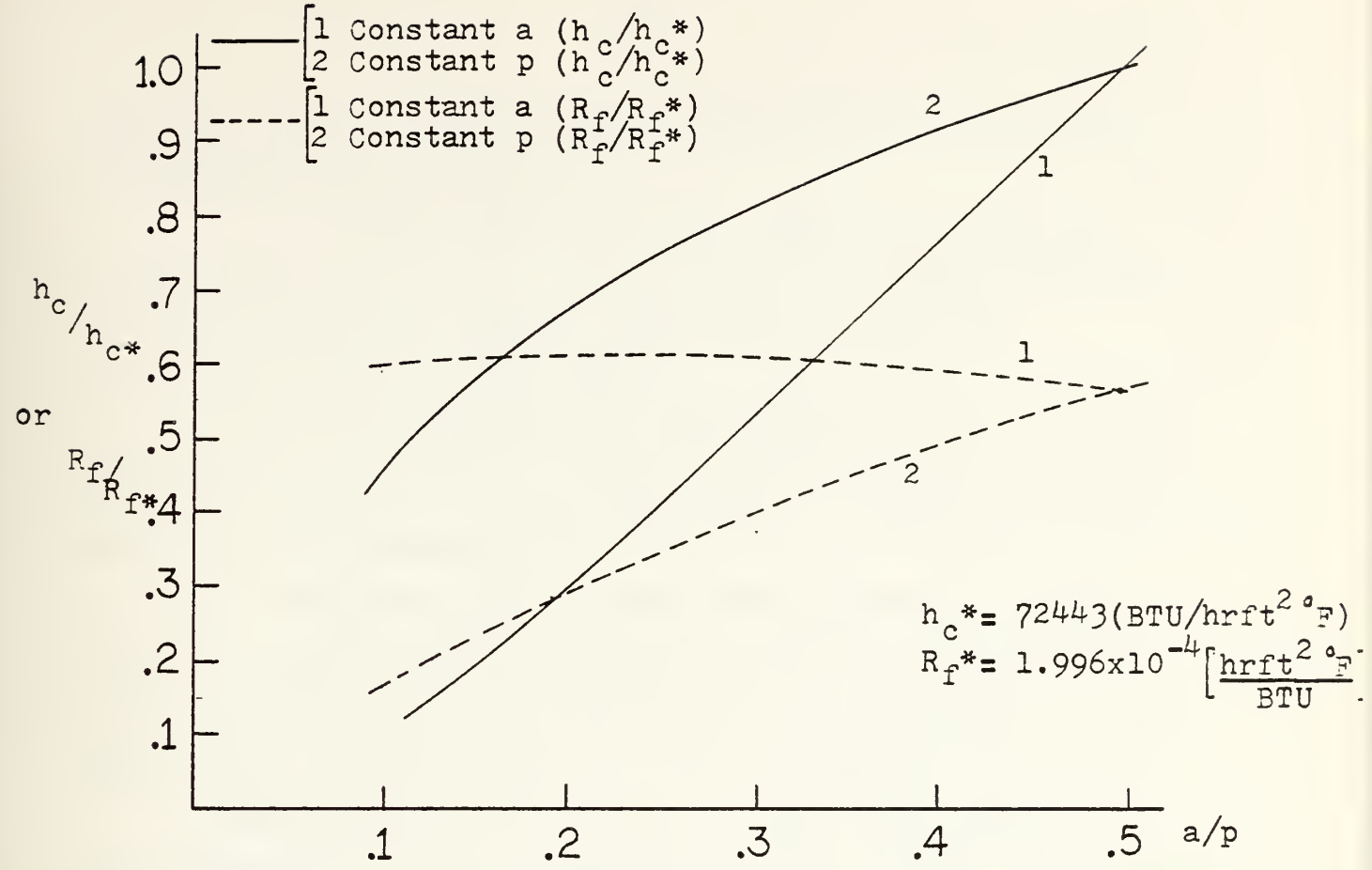



FIGURE 10 - Constant Amplitude (a=.0125in.) and Constant Pitch (p= .025in.) Curves

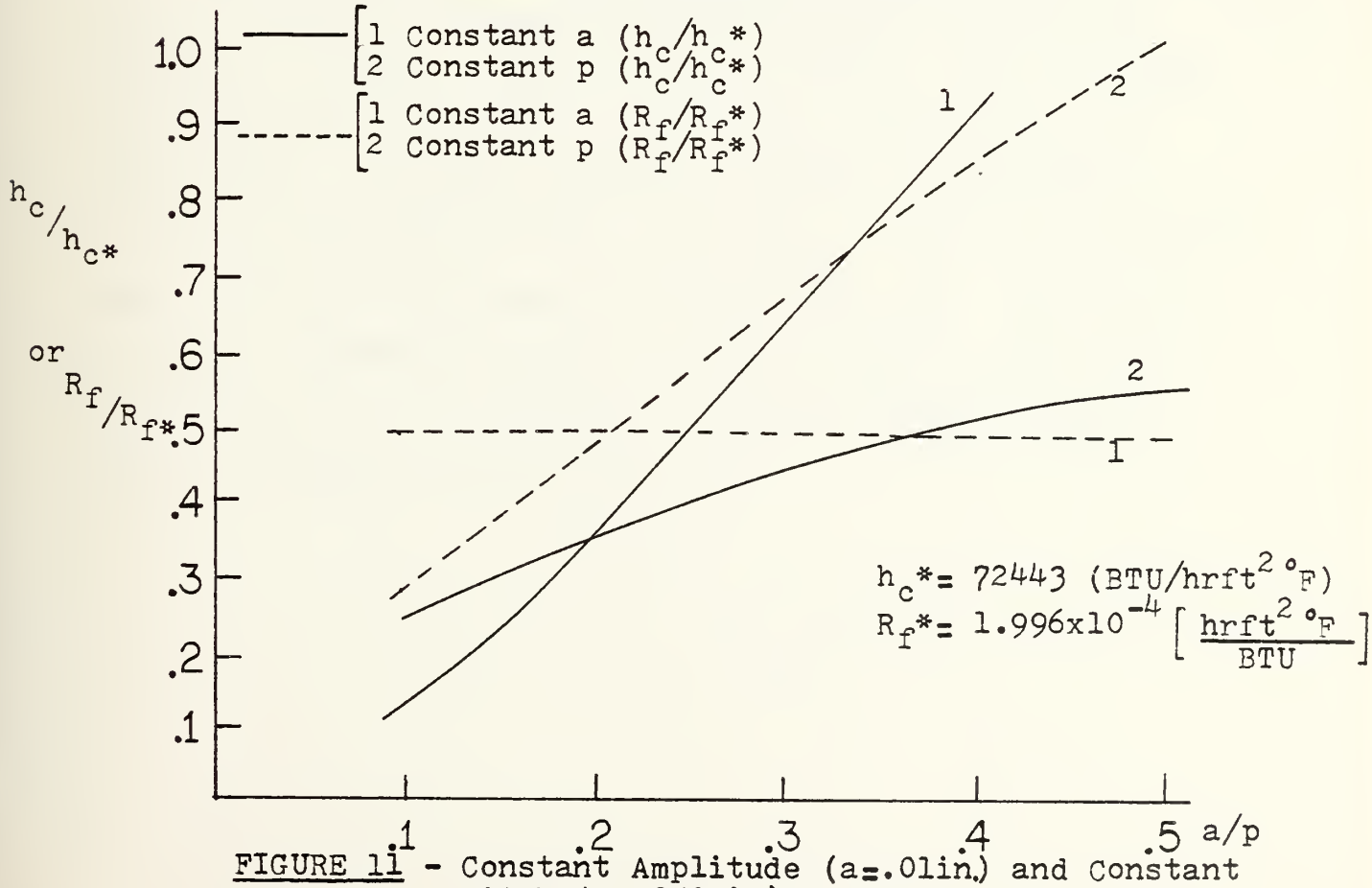



FIGURE 11 - Constant Amplitude (a=.01in.) and Constant Pitch (p=.050 in.) Curves

Equation (II-26). From Figures 10 and 11, the following conclusions can be inferred:

1. When comparing on the basis of flute pitch at the same a/p ratio, it can be seen that flute resistance is larger and heat transfer enhancement is less for increased values of flute pitch.

2. When comparing on the basis of flute amplitude at the same value of p, it can be seen that flute resistance is larger and heat transfer enhancement is greater for increased values of flute amplitude.

This indicates that the best tube geometry is small amplitude flutes with larger a/p ratios, and thus smaller values of flute pitch in comparison with other tubes. Thus the benefits of increased heat transfer enhancement are obtained with minimum increase in flute thermal resistance.

As demonstrated by Figure-8, greater heat transfer enhancement is achieved at smaller values of $W/W_F$, but there is a lower limit in practical condenser design for $W/W_F$. Reduced values of condensate drainage plate spacing are required to achieve smaller values of $W/W_F$. Close drainage plate spacing will complicate condenser fabrication and maintenance. Therefore acceptable values of condensate drainage plate spacing should be used as criteria for specifying $W/W_F$ in a practical condenser design. This criteria is used in the program VERTCON-2 to specify $W/W_F$. This effect of $W/W_F$ on condensate drainage plate spacing can also be observed in Figure-8, where the drainage plate spacing at Point A ($W/W_F = .25$) is equal to 1.632 ft. while at Point B ($W/W_F = .1$) the spacing is 0.622 ft.

## III. CONDENSATION IN THE PRESENCE OF NON-CONDENSABLE GASES

### Nomenclature

| | |
|---|---|
| $A$ | Flow area ($m^2$), ($ft^2$) |
| $a$ | Constant defined in Equation (III-6) |
| $B$ | Term defined by Equation (III-14a) |
| $C_{pf}$ | Specific heat of cooling water (J/kg °K), (BTU/lbm °F) |
| $D_o$ | External diameter of tube (m), (ft) |
| $D_n$ | Nominal diameter of axial fluted (m), (ft) tube |
| $D_p$ | Coefficient of vapor diffusion in gas (kg/m s)/Pa, (Lbm/ft s)/Lbf/ft$^2$ |
| $G_c$ | Vapor mass velocity approaching tube row (kg/$m^2$s), (Lbm/$ft^2$s) |
| $g$ | Gravitational acceleration ($m/s^2$), ($ft/s^2$) |
| $H_c$ | Specific enthalpy of vapor (J/kg), (BTU/Lb) |
| $h_{fg}$ | Specific latent heat (J/kg), (BTU/Lb) |
| $H_v$ | Specific enthalpy of vapor (J/kg), (BTU/Lb) |
| $L$ | Length of condenser (m), (ft) |
| $N$ | Number of tubes per row |
| $P_a$ | Partial pressure of air in bulk of mixture ($P_a$), ($lb/ft^2$) |
| $P_m$ | Pressure of mixture ($P_a$), ($lb/ft^2$) |
| $P_s$ | Partial pressure of vapor at condensate surface ($P_a$), ($lb/ft^2$) |
| $P_v$ | Partial pressure of vapor in bulk of mixture ($P_a$), ($lb/ft^2$) |
| $q$ | Heat flux ($W/m^2$), (BTU/hr$ft^2$) |
| $T_f$ | Temperature of coolant (°K), (°R, °F) |
| $T_{LM}$ | Mean temperature of coolant in tube (°K), (°R, °F) |
| $T_s$ | Temperature of condensate surface (°K), (°R, °F) |
| $T_v$ | Temperature of vapor (°K), (°R, °F) |
| $T_w$ | Temperature of outer tube wall (°K), (°R, °F) |
| $U_n$ | Heat transfer coefficient for convection and tube resistances as defined by Equation (II-6) (BTU/hr$ft^2$ °F) |
| $U_o$ | Overall heat transfer coefficient between vapor and coolant ($W/m^2$ °K), (BTU/hr$ft^2$ °F) |
| $V_v$ | Specific volume of vapor ($m^3$/kg), ($ft^3$/lb) |

$W_a$ Flowrate of air (kg/s), (lb/s)

$W_c$ Rate of condensation per row (kg/s), (lb/sec)

$W_v$ Flowrate of cooling fluid (kg/s), (lb/s)

$W_v$ Flowrate of vapor (kg/s), (lb/s)

$W_s$ Condenser steam flow (Kg/hr), (Lb/hr)

Subscripts

N Subscript indicating nominal tube diameter

v Subscript indicating vapor

1 Subscript indicating conditions at coolant inlet

2 Subscript indicating conditions at coolant outlet

Greek Symbols

$\varepsilon_v$ Volume ratio of air to mixture

$\varepsilon_w$ Weight ratio of air to mixture

$\varepsilon_{Wo}$ Weight ratio of air to mixture at outlet of air cooler

$\mu_a$ Absolute viscosity of air (n/m s)

$\mu_m$ Absolute viscosity of mixture (n/m s)

$\mu_v$ Absolute viscosity of vapor (n/m s)

$\xi$ Term defined by Equation (III- )

$\psi$ Coefficient of mass transfer $(kg/m^2s)/P_a$

Dimensionless Groups

$Nu_D$ Nusselt number defined by Equation (III-6)

$Re_m$ Reynolds number of vapor gas mixture

## HEAT AND MASS TRANSFER EQUATIONS

A significant decrease in condensing heat transfer coefficient ($h_c$) can result from the presence of very small amounts of non-condensable gases. The motion of the vapor with entrained gases moving towards a heat transfer surface, results in an accumulation of non-condensable gases. A vapor partial pressure gradient is required for the vapor to be able to diffuse through the gas blanket. This results in a liquid-vapor interface temperature below the temperature of the main vapor gas mixture. The heat and mass transfer equations developed by CHISHOLM (6) were applied in the VERTCON-2 condenser design program in predicting the condensation of steam in the presence of non-condensable gases. Diffusion coefficients obtained from experiments were used in this analysis by CHISHOLM (6).

The motion of the vapor towards the condenser surface is described by:

$$G_c = \psi(P_v - P_s) \qquad \text{(III-1)}$$

The rate of heat transfer per unit area is given by:

$$q = G_c\,(H_v - H_c) \qquad \text{(III-2)}$$

and, if its assumed that only laten heat is removed, then:

$$q = G_c h_{fg} \qquad \text{(III-3)}$$

From Clapeyron's approximate equations assuming that $P_v - P_s$ is small, then:

$$P_v - P_s = (T_v - T_s)\frac{1}{(T_v)V_v} \qquad \text{(III-4)}$$

By combining equations III-1, III-2, and III-4 results in the governing heat transfer equation within the vapor-gas mixture shown below:

$$q = \frac{\psi h_{fg}^2}{(T_v)V_v}(T_v - T_s) \qquad \text{(III-5)}$$

The coefficient of mass transfer is evaluated from:

$$Nu_D = aRe_m^{0.5}\xi^{-1/3}\varepsilon_v^{-0.6} \qquad \text{(III-6)}$$

where:

$$Nu_D = \frac{\psi D_o}{D_p} \qquad \text{(III-7)}$$

$$Re_m = \frac{W_v + W_a}{A} \quad \frac{D_o}{\mu_m} \qquad \text{(III-8)}$$

$$\xi = \frac{P_v - P_s}{P_m} \qquad \text{(III-9)}$$

For $Re_m > 350$

a = 0.52 for first tube row
= 0.67 for second tube row, and
= 0.82 for third and later tube rows

For $Re_m < 350$

a = 0.52 for all rows

The absolute viscosity of the mixture is determined from:

$$\mu_m = \left[(1-\varepsilon_v)\mu_v + 1.6.\varepsilon_v\mu_a\right]/(1+0.61\varepsilon_v) \qquad \text{(III-10)}$$

The coefficient of steam in air is obtained from:

$$D_p = \frac{2.93 \times 10^{-9}}{P_m} \left[\frac{T_v}{273}\right]^{0.8} \quad \text{(III-11)}$$

$$(kg/m\ s)/Pa$$

The ratio of air volume to total mixture volume is obtained from:

$$\varepsilon_v = P_a/P_m \quad \text{(III-12)}$$

and the partial pressures are evaluated from the equation:

$$\frac{P_m}{P_v} = 1 + 0.622 \frac{\varepsilon_w}{1-\varepsilon_w} \quad \text{(III-13)}$$

Combining equations III-5, III-6, III-7, and III-9 yields:

$$q = B(T_v - T_s)^{2/3} \quad \text{(III-14a.)}$$

where: $$B = \frac{D_p}{D_o} a \frac{Re_m^{0.5}}{\varepsilon_v^{0.6}} P_m^{1/3} \left[\frac{1}{T_v V_v}\right]^{2/3} h_{fg}^{5/3} \quad \text{III-14b.)}$$

The heat transfer coefficient between the condensate and the cooling water is evaluated from:

$$q = U(T_s - T_f) \quad \text{(III-15)}$$

where U is the overall heat transfer coefficient defined in equation (II- ).

Combining equations III-14 and III-15 yields:

$$\left[\frac{q}{B}\right]^{3/2} + \left[\frac{q}{U}\right] = T_v - T_f \quad \text{(III-16)}$$

or: $$q = (T_v - T_f) / \left[\frac{q^{\frac{1}{2}}}{B^{3/2}} + \frac{1}{U}\right] \quad \text{(III-17)}$$

For given values of B, and U and temperatures $T_v$ and $T_f$ this equation can be solved for q. Equation (III-17) will converge for any positive initial value of q. For the first tube row and first iteration assume q is zero and the coolant temperature is the inlet value, and for the next row and first iteration assume q and mean temperature of the coolant $T_{LM}$ from the previous row.

In evaluating U the condensate properties are first evaluated at the vapor temperature. After obtaining q on this basis, the overall heat-transfer coefficient between vapor and coolant is calculated from:

$$U_o = q/(T_v - T_L) \quad \text{(III-18)}$$

Using the overall heat transfer coefficient and log mean temperature difference for the coolant, Equation (III-18) can be written:

$$q = U_o \frac{(T_{f2} - T_{f1})}{\ln\left[\frac{T_v - T_{f1}}{T_v - T_{f2}}\right]} \quad \text{(III-19)}$$

substituting: $Q = W_f C_f (T_{f2} - T_{f1})$ (III-20)

results in:

$$\frac{T_v - T_{f1}}{T_v - T_{f2}} = \exp(-U_o \pi D_N LN / W_f C_{pf} \quad \text{(III-21)}$$

and the mean temperature of the coolant is evaluated from:

$$T_{LM} = T_v - (T_{f2} - T_{f1}) W_f C_f / U_o \pi D_N LN \quad \text{(III-22)}$$

For successive iterations $T_{LM}$ is used as the cooling fluid temperature ($T_f$) in Equation (III-17). At this point

the outer tube wall temperature and surface temperature of the condensate can be calculated, so that a more accurate value of U can be determined. This procedure is repeated with more accurate values of U, B, and $T_{LM}$ until successive values of q are within specified tolerances.

Then, the total condensation for a tube row is calculated from:

$$W_c = q\pi D_N LN/h_{fg} \qquad \text{(III-23)}$$

Standard design methods use either the BEAMA (17) code in the UK or the HEI (18) code in the US. Both codes recommend that the air ejector should be sized to reduce the vapor gas mixture leaving the condenser to 7.5 °F below the saturation temperature corresponding to condenser pressure. As an example a vacuum of 29 in.Hg corresponds to a ratio of non-condensable gas to mixture flowrate ($\varepsilon_{wo}$) to the ejector of:

$$\varepsilon_{wo} = \frac{W_A}{W_v + W_A} = 0.31$$

Figure-12 can be used to determine the flowrate of non-condensable gases ($W_A$) to be used in the sizing of the air ejector. In this study the BEAMA code was correlated using multiple linear regression (15), resulting in:

$$W_A = 10.0 + .1823\times10^{-3}W_s - .975\times10^{-10}W_s^2 \qquad \text{(III-24)}$$

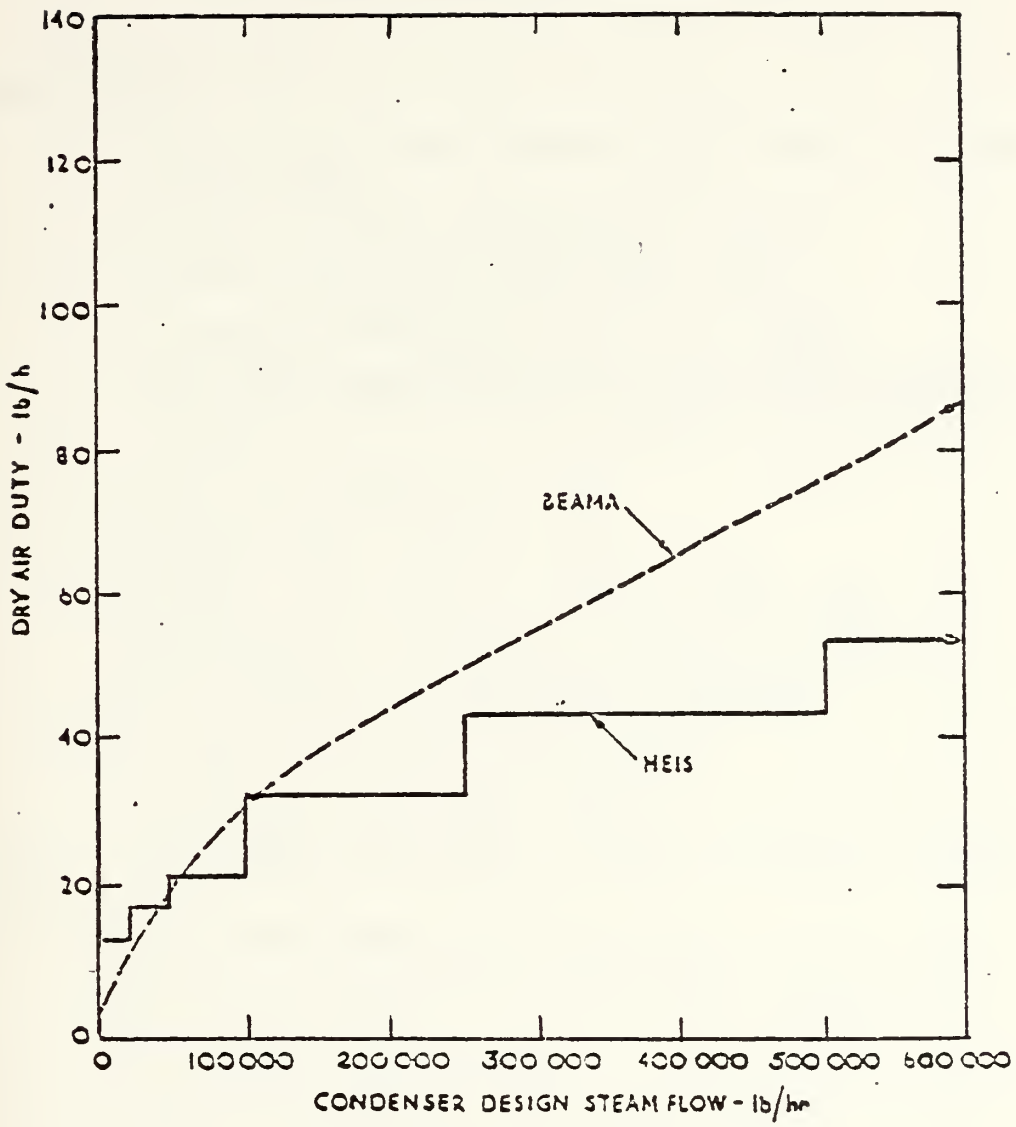



FIGURE 12

Recommended Air Extraction Capacity

## IV. PRESSURE DROP CALCULATIONS

Nomenclature

| | |
|---|---|
| $A_f$ | Minimum flow area between outer tubes of a tube bank ($ft^2$) |
| A | Constant defined by Equation (IV-1) |
| B | Constant defined by Equation (IV-1) |
| C | Conversion constant ($lb/ft^2$ to in.Hg) |
| $D_e$ | Equivalent diameter of flow area (the ratio of flow area to wetted perimeter multplied by four) (ft) |
| $D_i$ | Inside tube diameter (ft) |
| $D_o$ | Outside tube diameter (ft) |
| e | Internal tube helical ridge height (ft) |
| E | Dimensionless relative roughness = $\varepsilon/D_i$ |
| g | Gravitational acceleration ($ft/sec^2$) |
| L | Perimeter of tube bundle (ft) |
| $L_t$ | Condenser tube length (ft) |
| $L_1, L_2$ $L_3, L_4$ | Flow length along tube bundle perimeter defined in Figure-13 (ft) |
| $N_e$ | Equivalent number of tube rows |
| p | Pitch of internal tube helical ridging (ft) |
| P | Tube pitch (ft) |
| $P_t$ | Total pressure (in.Hg) |
| $P_s$ | Specified condenser pressure (static)(in.Hg) |
| $\Delta P_b$ | Pressure loss through tube bank (in.Hg) |
| $\Delta P_{bm}$ | Mean penetration pressure loss (in.Hg) |
| $\Delta P_{dm}$ | Mean total pressure loss around tube bundle (in.Hg) |
| $\Delta P_{en}$ | Entrance pressure loss (in.Hg) |
| $\Delta P_2, \Delta P_3$ $\Delta P_4$ | Pressure loss along flow length $L_2$, $L_3$, $L_4$ respectively (in.Hg) |
| $\Delta P_{WI}$ | Waterbox inlet loss (ft.of water) |

| | |
|---|---|
| $\Delta P_{WO}$ | Waterbox outlet loss (ft. of water) |
| $\Delta P_{TE}$ | Tube end loss (combined inlet and outlet), (ft. of water) |
| $V_f$ | Flow velocity of cooling water (ft/sec) |
| $V_s$ | Flow velocity of steam (ft/sec) |

Greek Symbols

| | |
|---|---|
| $\varepsilon$ | Tube surface roughness (ft) |
| $\rho$ | Steam density ($lb/ft^3$) |
| $\mu$ | Steam viscosity (lb/hr-ft) |

Dimensionless Groups

| | |
|---|---|
| f | Friction factor |
| $f^1$ | Friction factor defined by Equation (IV-19) |
| $f_{en}$ | Friction factor for internally enhanced tubes |
| Re | Reynolds number |

## IV.A TUBESIDE PRESSURE DROP

The total tubeside pressure drop is calculated from the summation of the following pressure drop components:

1. pressure loss in condenser tubes
2. tube end pressure loss (inlet and outlet)
3. waterbox pressure loss (inlet and outlet)

The pressure loss for smooth internal vertical tubes is evaluated using a friction factor relationship developed by CHURCHILL (19) as shown in Equation (IV-1) which is applicable for any flow regime as a function of Reynolds number and relative roughness of the tube.

$$f = 8.0\left[(8/Re)^{12} + 1.0/(A+B)^{3/2}\right]^{0.0833} \qquad \text{(IV-1)}$$

where:

$$A = \left[2.457\ \ln(1.0/(7/Pe)^{.08} + 0.27E)\right]^{16}$$

$$B = (37{,}530/Re)^{16}$$

$\epsilon$ = surface roughness

= $5 \times 10^{-6}$ ft. recommended value (20) for smooth tubing

$E = \epsilon/D_i$ dimensionless relative roughness where $D_i$ is the inside tube diameter (ft.)

Pressure loss for multiple-helix internal ridged tubes is calculated using friction factor ($f$ ) correlations evaluated from Equation (II-7) resulting from manufacturer's test data obtained from Reference-13. The friction factor data was correlated with $f_{en}$ as a function of Reynolds number, and the m & r operands with the following result:

$$f_{en} = 8\left[-\frac{1}{2.46\ \ln\left[r+(7/Re)^m\right]}\right]^2 \qquad \text{(IV-2)}$$

Tube properties and friction factor characteristics for multiple-helix internal ridged tubes can be found in Appendix-B.

Tube end and waterbox pressure losses for single and double pass condensers are calculated using pressure loss correlations evaluated from multiple non-linear regression using HEI standards (18). Pressure drop data from the HEI code can be found in Appendix-B. The resulting correlations are shown here:

Single Pass Condensers

Waterbox Inlet Loss:

$$P_{WI} = .01422V_f^{2.04} \qquad \text{(IV-3)}$$

Waterbox outlet Loss:

$$P_{WO} = .00237V_f^{2.33} \qquad \text{(IV-4)}$$

Tube End Loss (Combined Inlet & Outlet):

$$P_{TE} = .0178V_f^{2} \qquad \text{(IV-5)}$$

Double Pass Condensers

Waterbox Inlet Loss:

$$P_{WI} = .01422V_f^{2.04} \qquad \text{(IV-6)}$$

Waterbox Outlet Loss:

$$P_{WO} = .00635V_f^{2.1} \qquad \text{(IV-7)}$$

Tube End Loss (Combined Inlet & Outlet):

$$P_{TE} = .0356V_f^{2} \qquad \text{(IV-8)}$$

## IV.B SHELLSIDE PRESSURE DROP

Vapor condensing temperature is a function of vapor pressure, which follows a specific saturation temperature-pressure relationship. With the pressure drops encountered in the steam flow from the steam inlet to the air cooler section through the tube bundle, there is a resulting reduction in condensing temperature in the direction of flow. This temperature reduction will affect the rate of heat transfer in different sections of the condenser.

In Reference-6 HARRINGTON has noted that the condenser performance standards of the HEI (18) are based on static pressure, but that the condenser tube more closely senses a temperature related to the total or stagnation pressure. The method for evaluating the steam pressure distribution established by HARRINGTON along with the assumption that condenser performance is based on stagnation pressure was used in this study. Also, this can be considered as a one-dimensional design method where the vapor flow distribution is represented by a single-flow path. The flow path is considered as cross flow, radially inward from the tube bundle perimeter to the air cooler section of the condenser.

The pressure loss incurred by the steam flow can be considered as two components:

(1) steam distribution loss for steam flow from the inlet to around the tube bundle

(2) steam penetration loss for steam flow through the tube bundle from the tube bundle perimeter to the air cooler section of the condenser.

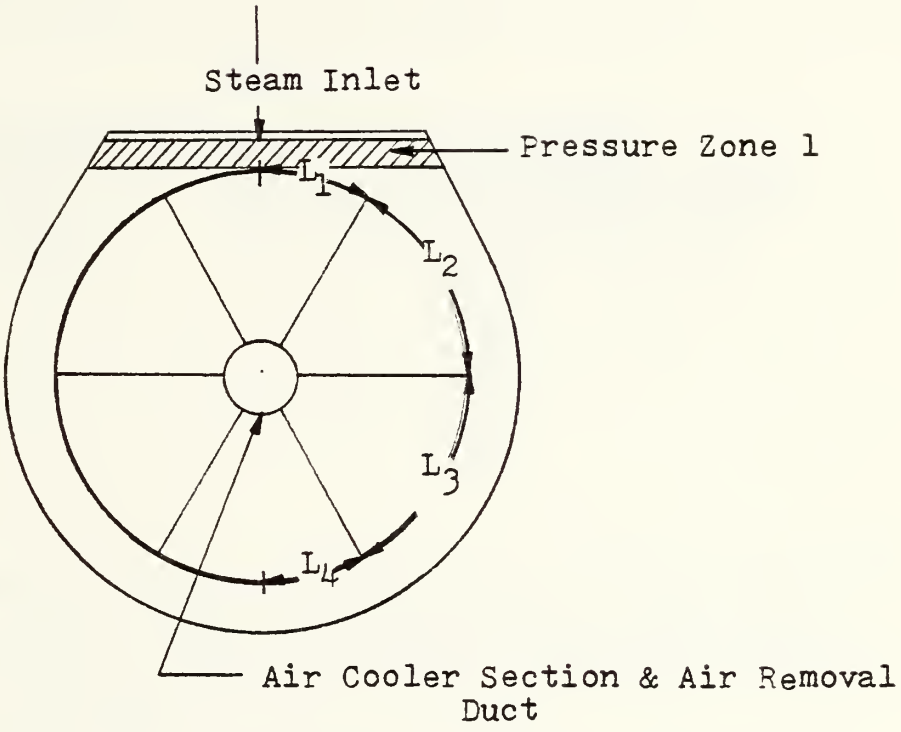



FIGURE 13

Steam Distribution Pressure Zones

Steam Distribution Loss

Refer to Figure-13for the following discussion. The total or stagnation pressure is calculated for pressure zone-1 and is evaluated from:

$$P_t = P_s + \frac{C \rho V_s^2}{2g} \qquad \text{(IV-9)}$$

The steam velocity required in Equation (IV-9) and for the entrance loss calculation is assumed to correspond to the recommended maximum main steam lane entrance velocities as shown in Table- 1 .

Table- 1

Recommended Steam Design Velocities

| Condenser Design Pressure (in.Hg) | Recommended Maximum Main Steam Lane Entrance Velocity, fps |
|---|---|
| 1 | 500 |
| 2 | 400 |
| 3 | 300 |
| 4 | 250 |
| 5 | 200 |

The entrance flow area will have to be sized accordingly with respect to this entrance flow velocity. All pressure losses around the tube bundle perimeter are based on the total pressure in zone-1.

The next step is to calculate the entrance pressure loss ($L_1$) from pressure zone-1 to the main steam inlet lane as follows:

$$\Delta P_{en} = 0.05 \frac{C \rho V_s^2}{2g} \qquad \text{(IV-10)}$$

The next step is to calculate pressure loss due to friction along $L_2$. Calculate the steam velocity from the

flow area in this zone. It assumed the net steam flow is reduced proportionately along the flow length $L_1$, $L_2$, $L_3$, and $L_4$.

Reynolds number can then be evaluated from:

$$Re = \frac{\rho D_e V_s}{\mu} \qquad \text{(IV-11)}$$

where $D_e$ is the equivalent diameter of flow area, which is the ratio of flow area to wetted perimeter multiplied by four.

Finally the pressure loss due to friction along $L_2$ is evaluated using the Tanning equation as follows:

$$P_{L2} = \frac{4C\rho f L_2 V_s^2}{D_e 2g} \qquad \text{(IV-12)}$$

The friction factor f is obtained from Figure-14 , or by evaluating the following correlation obtained by using multiple non-linear regression analysis (15), for an assumed roughness value.

$$f = .0436\ Re^{-.186} \qquad \text{(IV-13)}$$

This procedure is repeated to determine the frictional pressure loss for $L_3$ and $L_4$. This calculation scheme can be conducted with a greater number of smaller length segments to obtain a more accurate pressure distribution around the tube bundle.

The mean distribution pressure loss around the tube bundle can then be calculated as follows:

$$P_{dn} = \frac{L_1 P_{en} + L_2 P_2 + L_3 P_3 + L_4 P_4}{L_1 + L_2 + L_3 + L_4} \qquad \text{(IV-14)}$$

This mean total pressure identifies the condensing conditions at the tube bundle perimeter.

Steam Penetration Loss in Tube Bank

Calculation of Pressure loss through a tube bank is accomplished by replacing the actual tube bank with an equivalent rectangular bank that has the same perimeter and number of tubes. The equivalent number of rows in the equivalent tube bank depth is given by:

$$N_e = \frac{NP}{L} \qquad \text{(IV-15)}$$

Next calculate the velocity through the minimum flow area between the tubes. The flow area is obtained from:

$$A_f = \frac{L}{P}(P-D_o)L_T \qquad \text{(IV-16)}$$

The steam velocity entering the tube bank can be calculated using this value for flow area (A). Then Reynolds number is evaluated from:

$$Re = \frac{\rho D_o V_s}{\mu} \qquad \text{(IV-17)}$$

The terms ($\rho$ & $\mu$) in Equation (IV-17) are calculated at the temperature corresponding to the mean pressure at the tube bank entrance.

The tube bank pressure loss equation, which is modified for uniformly decreasing mass flow (21)

$$P_b = 4f^1(0.5 + \frac{Ne}{3})\frac{C\ V_s^2}{2g} \qquad \text{(IV-18)}$$

The friction factor $f^1$ is obtained from Figure-15, or by evaluating the following correlation obtained by applying multiple non-linear regression analysis (15).

$$f^1 = 65.5Re^{-.702}(P/D_o)^{-4.257} \quad (10 \leq Re \leq 1000) \qquad \text{(IV-19a.)}$$

$$f^1 = .591Re^{-.136}(P/D_o)^{-1.137} \quad (1000 < Re \leq 1.0x10^5) \qquad \text{(IV-19b)}$$

HARRINGTON (6) has suggested that the following equation be used to determine the mean tube bank pressure loss:

$$P_{bm} = 4f^1(0.5 + \frac{N_e}{4})\frac{C\rho V_s^2}{2g} \qquad \text{(IV-20)}$$

The mean condensing pressure in the tube bank is then evaluated from:

$$P_{bm} = P_t - \Delta P_{dm} - \Delta P_{bm} \qquad \text{(IV-21)}$$

All of the heat transfer calculations are based on the mean condensing pressure ($P_{bm}$) in the tube bank.

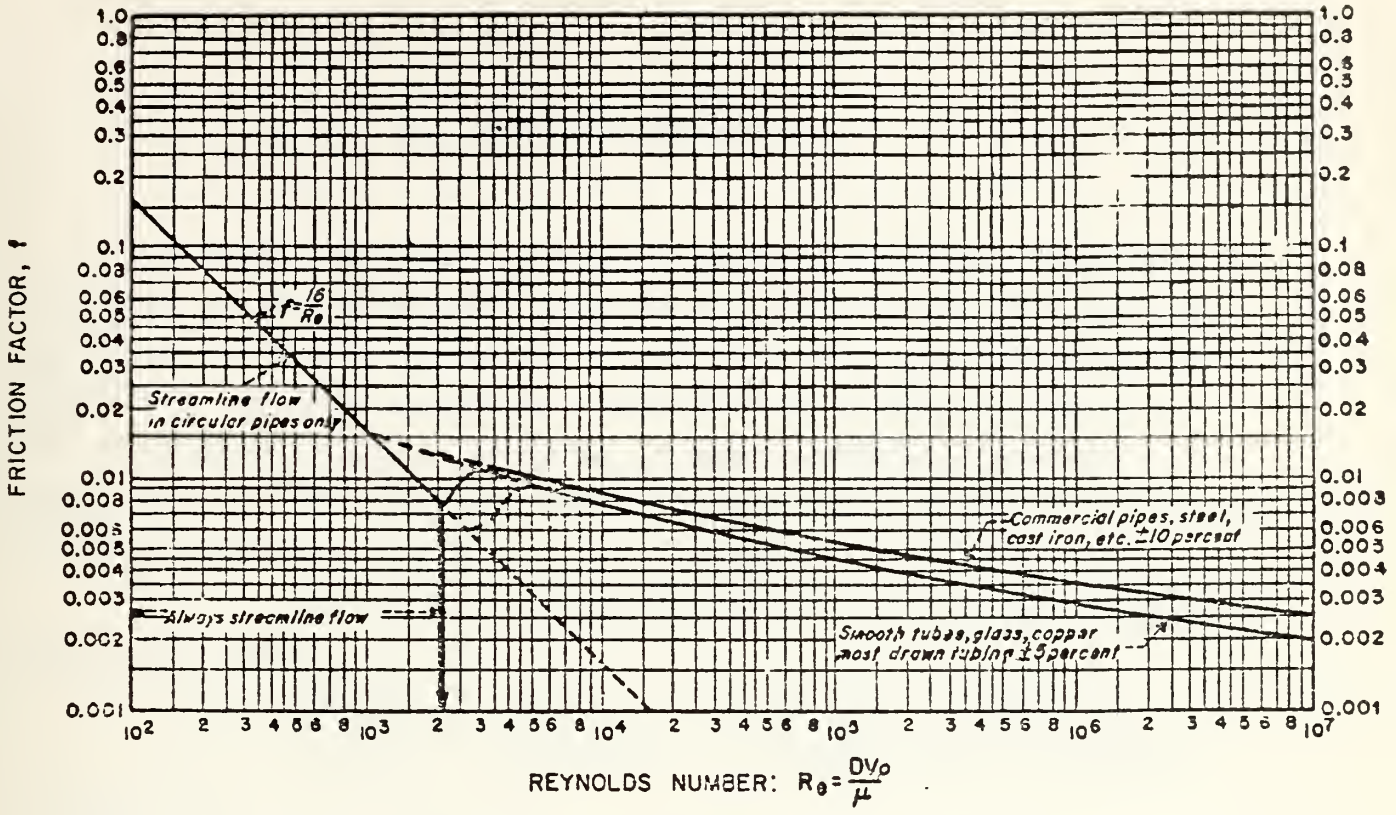

FRICTION FACTOR, f
f = 16/R_e
Streamline flow in circular pipes only
Always streamline flow
Commercial pipes, steel, cast iron, etc. ±10 percent
Smooth tubes, glass, copper most drawn tubing ±5 percent
REYNOLDS NUMBER: R_e = DV_o/μ

FIGURE 14

Friction Factor for Steam Distribution Losses

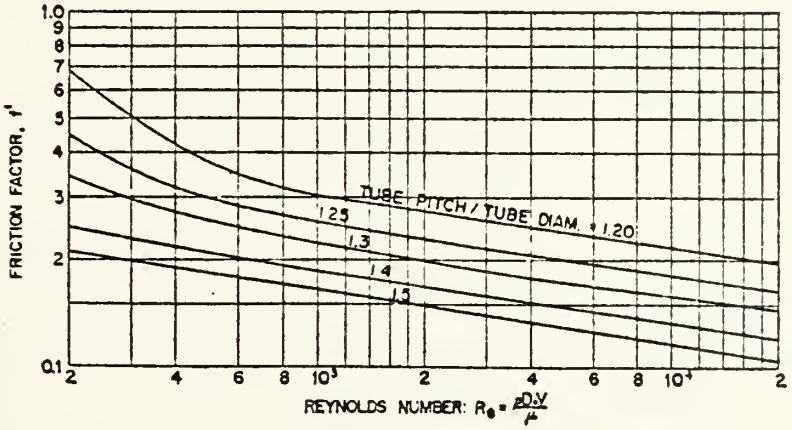



FIGURE 15

Friction Factor for Steam Penetration (Tube Bank) Losses

## V. COMPUTER MODELING

The basic procedure for heat enchanger sizing that was employed in the VERTCON-1 program (3) is used in the Vertical Condenser Sizing Program No. 2 (VERTCON-2). In addition the VERTCON-2 program considers condensation in the presence of non-condensable gases, and the program optimizes the condenser design for minimum weight and volume. Other features of the VERTCON-2 program are as follows:

1. The option of smooth internal tubes or internally enhanced tubes.
2. Design of single or double pass condensers.
3. Option of submarine, or surface ship condenser design.
4. Two basic configurations are available for double pass condensers:
   (a) Conventional return header design
   (b) "U-Tube" construction
5. Diagnostic messages used to warn when condenser drainage plate spacing is beyond specifications based on heat transfer performance and mechanical design.
6. Computation of recommended values for condenser drainage plate spacing and maximum main steam lane entrance velocity when they are not specified.

### Program Discription

The program VERTCON-2 is described here, and diagrammatically in the simplified program flow chart (Figure-16). The program is based on the FORTRAN-77 standard, and is listed in Appendix-A. Also included in Appendix-A are sample program input and output files.

The program input consists of input data files, and

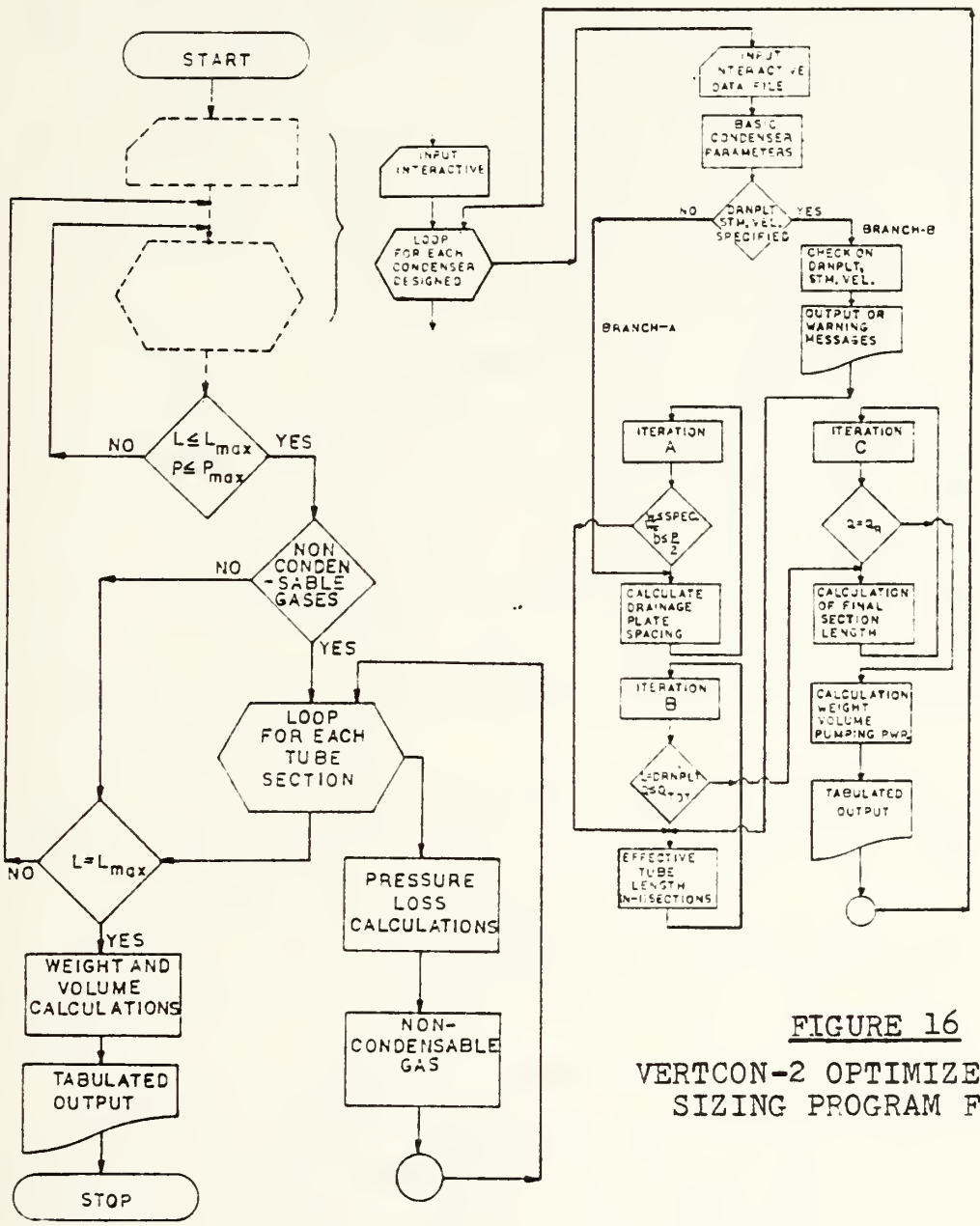



FIGURE 16

VERTCON-2 OPTIMIZED CONDENSER SIZING PROGRAM FLOWCHART

interactive input entered at the terminal. The input data file consists of tube geometry information for the various candidate tubes to be considered, and specification of condenser geometry, materials and operating conditions. The design features are selected interactively at the terminal. See Appendix-A for further details on data input.

The program is set up for sizing just one condenser or a series of condensers. After inputing data the program prompts at the terminal for selection of design features. If condenser drainage plate spacing and maximum main steam lane entrance velocity are not specified the program calculates recommended values (Branch-A and Iteration-A, Figure-16). The determination of these values are based on heat transfer performance and mechanical design to provide proper tube support. If these values are specified then the program checks if condenser drainage plate spacing is beyond recommended values (Branch-B, Figure-16). If the values specified exceed those dictated by proper design, then diagnostic warning messages are printed at the terminal and in the program output.

After input of interactive data at the terminal, the program conducts the selection of flow velocity and tube geometry to optimize the design as follows:

1. For the first candidate tube geometry the condenser is sized at the specified minimum flow velocity ($V = V_{min}$).

2.(a) If condenser length (L) is less than specified maximum condenser length ($L_{max}$) and pumping power is less than specified maximum pumping power ($P_{max}$) then Step 1 is repeated for a larger value of flow velocity.

2.(b) This process is repeated until $L = L_{max}$ or $P = P_{max}$.

3. If at $V_{min}$: L $L_{max}$ then this candidate tube is rejected because of insufficient enhancement.

4. Steps 1 through 3 are repeated for all candidate tubes.

The condenser sizing in Steps 1 through 4 is evaluated as follows:

1. In iteration B and C (Figure-16) the total effective tube length for the condenser is calculated by adding N tube sections of length equal to the drainage plate spacing until a proper heat balance is obtained. Iteration B adds up N-1 of these sections of equal length. Iteration C then calculates the final section length, which may have a length less than the specified drainage plate spacing.

2. After the total effective tube length is determined the condenser geometry, weight, volume, and system power are calculated for each condenser.

It should be noted that the influence of non-condensable gases has not been considered yet, and a multiplicative factor ($F_{NC}$) is simply used to account for non-condensable gas effects.

At this point the minimum condenser sizing for each tube configuration are compared to select the condenser configuration with overall minimum weight and volume.

If the option is selected, the program will now consider the effects of non-condensable gases. Using the tube bundle and shell geometry of the selected condenser configuration, the mean pressure around the tube bundle is determined. With the conditions identified at the tube bundle perimeter the program steps through the condenser tube bundle row by row to calculate the effective tube length at local conditions including non-condensable gas effects.

If the condenser length calculated from the consideration of non-condensable gases differs from the original calculated value, then a new value for the multiplicative factor $F_{NC}$ is selected and the complete process including tube and flow velocity selection is repeated until these length values

are equal. By conducting the calculations in this manner the non-condensable gas effects will be evaluated a minimum number of times, thus significantly reducing computer solution time. There is a limit to the number ($N_I$) of these iterations that will be completed by the program. If after $N_I$ iterations there has not been proper convergence, then warning messages are displayed on the terminal screen and printed in the program output. This indicates that one of the specified condenser parameters is out of limits for the particular condenser design. Non-convergence would most likely be due to factors that influence steam-side heat transfer performance. One culprit could possibly be too small a value for tube pitch-to-diameter ratio, that would cause excessive steam side pressure drop.

The tabulated program output displays for each condenser the following information:

(1) All input data and design features selected.

(2) If applicable, diagnostic warning messages.

(3) Detailed weight breakdown of condenser components.

(4) Total condenser volume, center of gravity, and total weight in dry and wet conditions, and system pumping power.

# APPENDIX A

## DATA INPUT

See Table-3 for a sample input data file

## INPUT DATA FILE

1. Assigned Logical Operator - 20 to specify input device. See lines 47-52 of progress listing for further details.
2. Free Format
3. Variable names starting with I, J, K, L, M, or N are integer variables and others are real variables, except for specific cases as noted.

## LINE 1

Enter: SUB or SUR

SUB - Submarine Condenser Design

SUR - Surface Ship Condenser Design

## LINE 2

Enter: NUMBT,LMAX, PMAX, VMIN, VMAX

NUMBT- Number of Tube configurations to be tested

LMAX - Maximum condenser length (ft)

PMAX - Maximum pumping power (hp)

VMIN - Minimum coolant flow velocity (ft/sec)

VMAX - Maximum coolant flow velocity (ft/sec)

Note: LMAX is a real variable

## LINE 3

Enter: TCI, TSAT, QSTM, STMLD, PSAT PD

TCI - Inlet Coolant Temperature ( F)

TSAT - Steam Inlet Saturation Temperature ( F)

QSTM - Quality of Inlet Steam to Condenser

STMLD - Steam Condenser (lb/hr)

TABLE-2: SAMPLE INPUT DATA FILE
..4 TUBE TEST CASE..

```
SUB
4,24.34,275.,8.,15.
66.1,143.89,.85,257000.,6.5,1.35
2.,6.,.625,.5625,.5625,2.64,488.
558.,488.,488.,488.,558.,282.,1.56E7,.56,488.
1,.555,.625,.01,.245,0.,0.
48.,0.,0.,9.6,.00033
2,.555,.625,.01,.245,.0204,.191
48.,.59,.00197,9.6,.00033
2,.555,.625,.01,.245,.0125,.4750
48.,.762,0.0,9.6,.00033
2,.555,.625,.01,.245,.024,.0949
48.,.58,.0075,9.6,.00033
```

PSAT - Condenser Operating Pressure (in.Hg) Absolute
PD - Tube pitch-to-diameter ratio

LINE 4

Enter: TIS, TOTS, TSHL, TTSP, THW, THDR, ITSD

TIS - Thickness of internal tube sheet (in.)
TOTS - Thickness Outer Tube Sheet (in)
TSHL - Thickness of condenser shell (in.)
TTSP - Thickness tube support plate (in.)
THW - Thickness of hot well plate (in.)
THDR - Thickness of Headers (in.)
ITSD - Internal tube sheet density ($lb/Ft^3$)(real variable)

Note: ITSD is a real variable.

LINE 5

Enter: OTSD, SHLD, TSPD, HWD, HDRD, TBD, ETUBE, TCOVER, COVERD

OTSD - Outer tube sheet density ($lb/ft^3$)
SHLD - Shell density ($lb/ft^3$)
TSPD - Tube support plate density ($lb/ft^3$)
HWD - Hotwell plate density ($lb/ft^3$)
HDRD - Header density ($lb/ft^3$)
TBD - Tube bundle density ($lb/ft^3$)
ETUBE - Modulus of elasticity of tube material ($lb/in^2$)
TCOVER - Thickness of U-tube header cover (in.)
COVERD - Density of U-tube header cover (in.)

LINE 6

Enter: NTYPE, DI, DW, AE, AEPE, EI, PI

NTYPE - Tube type: a. for smooth internal tube
b. for doubly enhanced tubes
DI - Internal tube diameter (in.)
DW - Tube wall diameter (in.)
AE - External flute amplitude (in.)
AEPE - Amplitude-to-pitch ratio of external ridging
EI - Internal helix ridge height (in.)

PI - Pitch of Internal Helix (in.)

Note: For smooth internal tube (NTYPE-1) enter ØØ for EI, PI. Values of EI and PI are obtained from tube data in Appendix-B.

LINE 7

Enter: FLUTE, MI, RI, KW, RSCALE

FLUTE - Number of flutes per tube

MI - Operand for friction factor, helical internal ridging (real variable)

RI - Operand for Friction Factor, Helical internal ridging

KW - Conductivity of tube wall (BTU/hr-ft- F)

RSCALE - Fouling resistance (hr-ft- F/BTU)

Note: MI is a real variable.

To consider more tube configurations, repeat lines 6 and 7 for each additional tube to be included in the analysis.

INTERACTIVE DATA INPUT

Enter data at the terminal as follows:

When the program commences, the following message appears on the terminal screen:

** WELCOME TO PROGRAM: VERTCON, VERSION-2 **

ENTER PROGRAM RUN NUMBER

Enter Run number, which can be an integer from 0-to-999.

Next at the terminal appears:

ENTER THE NUMBER OF CONDENSERS TO BE SIZED

Enter the number of condensers to be sized, which is also an integer from 1-to-999.

Next at the terminal appears:

ENTER "1" FOR SINGLE PASS, OR "2" FOR DOUBLE PASS CONDENSER

Enter the proper integer number 1 or 2 for single or double pass condenser.

Next at the terminal appears:

TWO BASIC CONFIGURATIONS ARE AVAILABLE FOR DOUBLE PASS CONDENSERS:

(1) CONVENTIONAL RETURN HEADER DESIGN
(2) "U-TUBE" CONSTRUCTION

ENTER 1 OR 2 FOR CONFIGURATION SELECTION

Enter the proper integer number 1 or 2 for configuration selection.

Next at the terminal appears:

ENTER: (1) COOLANT FLOW RATE (GPM)

Enter as real number with units of gallons per minute.

Next at the terminal appears:

DO YOU WANT TO INCLUDE IN THIS ANALYSIS THE EFFECTS OF CONDENSATION IN THE PRESENCE OF NON-CONDENSABLE GASES YES OR NO

Enter YES or Y, NO or N for proper choice.

Next at the terminal appears:

DO YOU WANT TO SPECIFY MAXIMUM MAIN STEAM LANE ENTRANCE VELOCITY (FT/SEC)?

YES OR NO

Enter YES or Y, NO or N for proper choice.

If yes is entered, Next at the terminal appears:

ENTER MAXIMUM MAIN STEAM LANE ENTRANCE VELOCITY (FT/SEC)

Enter this as a real number.

If no is entered, Next at the terminal appears:

PROGRAM WILL SELECT RECOMMENDED MAXIMUM MAIN STEAM LANE ENTRANCE VELOCITY USING CONDENSER OPERATING PRESSURE AS SELECTION CRITERIA.

Next at the terminal appears:

DO YOU WANT TO SPECIFY CONDENSATE DRAINAGE PLATE SPACING (FT.)?

YES OR NO

Enter YES or Y, NO or N for proper choice.

If yes is entered, Next at the terminal appears:

ENTER CONDENSATE DRAINAGE PLATE SPACING (FT.)

Enter this as a real number.

If no was entered, Next at the terminal appears:

PROGRAM WILL SELECT RECOMMENDED CONDENSATE
DRAINAGE PLATE SPACING

Note: At this point, entry of Interactive Data is completed for the first condenser. The program now completes the sizing for the first condenser and will prompt the terminal for further interactive data for any additional condensers.

. . . TABLE-3: SAMPLE OUTPUT DATA FILE . . .

** PROGRAM: VERTCON,VERSION-2 **

PROGRAM RUN NUMBER( 1)



**** VARIABLE LIST ****

AE= EXTERNAL FLUTE AMPLITUDE (IN.)
AEPE= AMPLITUDE-TO-PITCH RATIO OF EXTERNAL RIDGING
ATB= AREA OF TUBE BUNDLE (FT**2)
ATS= AREA OF TUBE SHEET (FT**2)
DI= INTERNAL TUBE DIAMETER (IN.)
DPTHDR= HEADER DEPTH (FT.)
DRYCG= HEIGHT OF CENTER OF GRAVITY ABOVE CONDENSER BOTTOM AT DRY WEIGHT (FT.)
DTB= DIAMETER OF TUBE BUNDLE (FT.)
DTS= DIAMETER OF TUBE SHEET(FT.)
DW= TUBE WALL DIAMETER (IN.)
EI= OPERAND FOR FRICTION FACTOR HELICAL INTERNAL RIDGING
FE= OPERAND FOR CONDENSING HEAT TRANSFER COEFFICIENT
KW= CONDUCTIVITY OF TUBE WALL
MI= OPERAND FOR FRICTION FACTOR; HELICAL INTERNAL RIDGING
NTYPE= TUBE TYPE: 1-FOR SMOOTH INTERNAL TUBE; AND 2-FOR DOUBLY ENHANCED TUBES
PI= PITCH OF INTERNAL HELIX(IN.)
PSAT= CONDENSER OPERATING PRESSURE (IN.HG)ABSOLUTE
QSTM= QUALITY OF INLET STEAM TO CONDENSER
RI= OPERAND FOR FRICTION FACTOR, HELICAL INTERNAL RIDGING
RSCALE= FOULING RESISTANCE (HR-FT**2-DEG.F/BTU)
STMLD= STEAM LOAD (LBM/HR)
TSAT= SATURATION TEMP. (DEG.F)
VTB= VOLUME OF TUBE BUNDLE (FT**3)
WEXP= EXPANSION JOINT WEIGHT (LB.)

(BTU/HR-FT-DEG.F)
LANE= STEAM LANE BREADTH (FT.)
LHW= LENGTH OF HOTWELL (FT.)
LTOT= TOTAL TUBE LENGTH (FT.)
TCI= INLET COOLANT TEMP. (DEG.F)
WMISC= WEIGHT OF MISCELLANEOUS COMPONENTS (LB.)
WETCG= HEIGHT OF CENTER OF GRAVITY ABOVE CONDENSER BOTTOM AT WET WEIGHT (FT.)

```
                    ******************************
                        CONDENSER NUMBER(  1)
                    ******************************
. . CONDENSER TYPE: SUBMARINE DESIGN
. . DOUBLE PASS CONDENSER WITH CONVENTIONAL RETURN HEADER DESIGN.
. . DOUBLY ENHANCED TUBES . .
. . TUBE NUMBER(  4) SELECTED . .
                          **** DATA INPUT ****
. . FLOW RATE:  7900. (GPM)  . . FLOW VELOCITY: 10.22 (FT/SEC) . .
DI = 0.555    DW = 0.625    AE = 0.010    AEPE = 0.2450    NTYPE =  2
EI = 0.0240   PI = 0.0949   FE =  4.54    MI = 0.580       RI = 0.0075
KW =   9.60   TCI =  66.10  PSAT = 6.50   QSTM = 0.85      TSAT =143.89
STMLD =  257000.0           RSCALE = 0.00033

               **** DATA OUTPUT - CONDENSER SIZING ROUTINE ****
. . MAXIMUM MAIN STEAM LANE ENTRANCE VELOCITY: 200. (FT/SEC)
. . CONDENSATE DRAINAGE PLATE SPACING:  1.06 (FT.)

. . CONDENSER HEAT LOAD:    0.22094E+09 (BTU/HR)
. . OUTLET COOLANT TEMP:    123.44 (DEG.F)
. . TOTAL NUMBER OF TUBES:  2050.0

. . EFFECTIVE TUBE LENGTH:           14.39 (FT.)
. . AVG. HEAT TRNFR. COEFF. C.W.:    6096. (BTU/HR-FT**2-DEG.F)
. . AVG. HEAT TRNFR. COEFF. COND.:  41488. (BTU/HR-FT**2-DEG.F)
. . AVG. OVERALL HEAT TRNFR. COEFF.:  994. (BTU/HR-FT**2-DEG.F)

LTOT = 16.24     LANE =  1.24     DTB =   3.68    ATB =  10.66
VTB = 153.39     DTS =  6.16      ATS =  29.81


          **** DATA OUTPUT - WEIGHT & VOLUME CALCULATIONS ****

WEXP =   440.1   WMISC = 11432.2   LHW =  5.02     DPTHDR =  3.08
```

| | MATERIAL DENSITY(LB/FT**3) | THICKNESS (IN.) | WEIGHT (LB.) |
|---|---|---|---|
| OUTER TUBE SHEET----- | 558.0 | 6.0000 | 13879. |
| INNER TUBE SHEET----- | 488.0 | 2.0000 | 4046. |
| TUBE SUPPORT PLATE--- | 488.0 | 0.5625 | 2040. |
| TUBE BUNDLE---------- | 282.0 | *.**** | 5531. |
| CONDENSER SHELL------ | 488.0 | 0.6250 | 8059. |
| HOTWELL-------------- | 488.0 | 0.5625 | 3569. |
| WATERBOX------------- | 558.0 | 2.6400 | 8163. |

. . TOTAL CONDENSER DRY WEIGHT= 57160.9 (LB.)
. . TOTAL CONDENSER WET WEIGHT= 65432.0 (LB.)
29.21 (TON)

. . TOTAL CONDENSER HEIGHT= 24.34 (FT.)
. . OUTER SHELL DIAMETER= 6.27 (FT.)
. . ENCLOSED BOX VOLUME= 1023.6 (FT**3)
DRYCG = 12.28 WETCG = 11.69

. . CONDENSER FRICTIONAL HEAD LOSS= 35.8 (FT.)
. . TOTAL SYSTEM HEAD LOSS= 88.8 (FT.)
. . TOTAL SYSTEM PUMPING POWER= 246.8 (HP)

```
C**********************
C  PROGRAM: VERTCON2
C**********************
C PROGRAM "VERTCON2" IS A PRELIMINARY DESIGN ROUTINE USED TO OPTIMIZE
C VERTICAL TUBE STEAM SHIP/SUBMARINE PROPULSION CONDENSERS TO PRODUCE
C CONDENSER DESIGNS WITH MINIMUM WEIGHT AND VOLUME. THIS PROGRAM CAN
C ALSO  CONSIDER CONDENSATION IN THE PRESENCE OF NON-CONDENSABLE GASES.
C THE PROGRAM TAKES INITIAL INPUT DATA BOTH INTERACTIVELY AND FROM DATA
C FILES. THE PROGRAM OUTPUT CONSISTS OF TABULATED VALUES FOR TOTAL &
C COMPONENT CONDENSER WEIGHTS, VOLUMES, AND PUMPING POWER.
C
C TWO BASIC VERTICAL TUBE CONFIGURATIONS ARE AVAILABLE WITH THIS PROGRAM
C AND THEY ARE: (1) DOUBLY ENHANCED TUBES, OR (2) SMOOTH INTERNAL TUBES
C WITH EXTERNAL ENHANCEMENT ONLY. SEVERAL GEOMETRIC VARIATIONS WITHIN
C THESE TWO BASIC CONFIGURATIONS CAN BE INVESTIGATED WITH "VERTCON1".
C
C ALSO, TWO BASIC CONSTRUCTION CONFIGURATIONS ARE AVAILABLE FOR "DOUBLE-
C PASS" CONDENSERS: (1) CONVENTIONAL RETURN HEADER DESIGN, AND
C                   (2) "U-TUBE" CONSTRUCTION
C
C THE VALUES OF CONDENSATE DRAINAGE PLATE SPACING (FT.), AND MAXIMUM
C MAIN STEAM LANE ENTRANCE VELOCITY (FT/SEC) MAY BE SPECIFIED BY THE
C USER INTERACTIVELY WHEN RUNNING THIS PROGRAM. IF THESE VALUES ARE
C NOT SPECIFIED, THEN THE PROGRAM WILL AUTOMATICALLY DETERMINE
C RECOMMENDED VALUES.
C
      DIMENSION DATA(500,5), OUTPUT(600,6),HWAVG(100),HCAVG(100),
     1  UAVG(100),TLT(100),XLTOT(100),XVM(100),TDATA(100,13),EFF(100)
      REAL ITSD,KSAT,KSW,KW,L1,LANE,LHW,LTOT,MI,MU,NT,NU,LMIN,LMAX,
     1  LMIN1,LMAX1,LMIN2,LMAX2,LPLUS,MULT1,MULT2,MULT3,MULT4,MULTP,MF
      INTEGER STM,CONFIG,CTYPE,DRAIN,GAS,OPT
      COMMON STMLD,TSAT,PSAT,STM,TOTRAD,ITOT,LANE,QSTM,PD,EFFL,
     1       TOTL,QT,AVGU,RADIUS(250),NTUBES(250)
      DATA NOE/'NO  '/,NO/'N   '/,ITYPE/'SUB '/
      DATA PIE/3.1415927/
```

```
C
C * * * STATEMENT FUNCTIONS FOR THERMAL & MECHANICAL PROPERTIES * * *
C
      NU(TCS)= .122181 - .1481615E-02 * TCS + .5516445E-05 * TCS**2
      RE(VM)=    VM * DI * 300. / NU(TCS)
      PRSW(TCS)= .0014369 * TCS**2 - .3134 * TCS + 21.89
      KSW(TCS)=  -.000003086 * TCS**2 + .001 * TCS + .291
C NU=   KINEMATIC VISCOSITY OF COOLING WATER (FT**2/HR)
C RE=   REYNOLDS NUMBER COOLING WATER
C PRSW= PRANDTL NUMBER COOLING WATER
C KSW=  CONDUCTIVITY OF COOLING WATER (BTU/HR-FT-DEG.F)
C
      FF(ARG2)= -1. / (2.46 * LOG(ARG2**MI + RI))
      ST(ARG3,ARG4)= FF(ARG2) / (5.68 * ARG3**(-.125) *SQRT(PRSW(TCS))
     1  * ARG4**.136 + GAMMA)
C FF= FRICTION FACTOR [SQRT(F/8)]
C ST= STANTON NUMBER
C
      KIN= 20
      KOUT= 21
      KSCR= 6
C KIN,KOUT, AND KSCR ARE OPERATORS THAT SPECIFY INPUT & OUTPUT DEVICES.
C IN THIS CASE KIN & KOUT SPECIFY INPUT & OUTPUT DATA FILES RESPECTIVELY
C AND KSCR SPECIFIES INPUT & OUTPUT AT COMPUTER TERMINAL.
C
C * * * * DATA(INTERACTIVE & FILE)-INPUT/OUTPUT * * * *
C
      WRITE(KSCR,100)
 100  FORMAT(1X,T14,'* * WELCOME TO PROGRAM: VERTCON,VERSION-2 * *'//T
     12,'ENTER PROGRAM RUN NUMBER')
      READ(KSCR,*) NUMBER
      WRITE(KOUT,102) NUMBER
 102  FORMAT(/////T21,'** PROGRAM: VERTCON,VERSION-2 **'////T25,'PROGR
     1AM RUN NUMBER(',I3,')'//////)
C
      WRITE(KOUT,104)
```

```
104   FORMAT(1X,T26,'**** VARIABLE LIST ****'//T2,'AE= EXTERNAL FLUTE A
     1MPLITUDE (IN.)',T38,'.',T41,'MI= OPERAND FOR FRICTION FACTOR;'/T2,
     1'AEPE= AMPLITUDE-TO-PITCH RATIO OF',T38,'.',T45,'HELICAL INTERNAL
     1RIDGING'/T6,'EXTERNAL RIDGING',T38,'.',T41,'NTYPE= TUBE TYPE: 1-FO
     1R SMOOTH IN-'/T2,'ATB= AREA OF TUBE BUNDLE (FT**2)',T38,'.',T45,'T
     1ERNAL TUBE; AND 2-FOR DOUBLY'/T2,'ATS= AREA OF TUBE SHEET (FT**2)'
     1,T38,'.',T45,'ENHANCED TUBES'/T2,'DI= INTERNAL TUBE DIAMETER (IN.)
     1',T38,'.',T41,'PI= PITCH OF INTERNAL HELIX(IN.)'/T2,'DPTHDR= HEADE
     1R DEPTH (FT.)',T38,'.',T41,'PSAT= CONDENSER OPERATING PRESSURE'/T2
     1,'DRYCG= HEIGHT OF CENTER OF GRAVITY',T38,'.',T45,'(IN.HG)ABSOLUTE
     1'/T6,'ABOVE CONDENSER BOTTOM AT DRY',T38,'.',T41,'QSTM= QUALITY OF
     1 INLET STEAM TO'/T6,'WEIGHT (FT.)',T38,'.',T45,'CONDENSER'/T2,'DTB
     1= DIAMETER OF TUBE BUNDLE (FT.)',T38,'.',T41,'RI= OPERAND FOR FRIC
     1TION FACTOR,'/T2,'DTS= DIAMETER OF TUBE SHEET(FT.)',T38,'.',T45,'H
     1ELICAL INTERNAL RIDGING')
        WRITE(KOUT,106)
106     FORMAT(1X,T2,'DW= TUBE WALL DIAMETER (IN.)',T38,'.',T41,'RSCALE=
     1 FOULING RESISTANCE'/T2,'EI= OPERAND FOR FRICTION FACTOR',T38,'.',
     1T45,'(HR-FT**2-DEG.F/BTU)'/T6,'HELICAL INTERNAL RIDGING',T38,'.',T
     141,'STMLD= STEAM LOAD (LBM/HR)'/T2,'FE= OPERAND FOR CONDENSING HEA
     1T',T38,'.',T41,'TSAT= SATURATION TEMP. (DEG.F)'/T6,'TRANSFER COEFF
     1ICIENT',T38,'.',T41,'VTB= VOLUME OF TUBE BUNDLE (FT**3)'/T2,'KW= C
     1ONDUCTIVITY OF TUBE WALL',T38,'.',T41,'WEXP= EXPANSION JOINT WEIGH
     1T (LB.)'/T6,'(BTU/HR-FT-DEG.F)',T38,'.',T41,'WMISC= WEIGHT OF MISC
     1ELLANEOUS'/T2,'LANE= STEAM LANE BREADTH (FT.)',T38,'.',T45,'COMPON
     1ENTS (LB.)'/T2,'LHW= LENGTH OF HOTWELL (FT.)',T38,'.',T41,'WETCG=
     1HEIGHT OF CENTER OF GRAVITY'/T2,'LTOT= TOTAL TUBE LENGTH (FT.)',T3
     18,'.',T45,'ABOVE CONDENSER BOTTOM AT WET'/T2,'TCI= INLET COOLANT T
     1EMP. (DEG.F)',T38,'.',T45,'WEIGHT (FT.)'/)
C
        WRITE(KSCR,108)
108     FORMAT(1X,'ENTER THE NUMBER OF CONDENSERS TO BE SIZED')
        READ(KSCR,*) NCOND
C
        DO 10 N= 1, NCOND
C
```

```
      CONFIG= 3
      WRITE(KOUT,110)
 110  FORMAT('1',T21,'******************************')
      WRITE(KOUT,112) N
      WRITE(KSCR,112) N
 112  FORMAT(1X,T25,'CONDENSER NUMBER(',I3,')')
      WRITE(KOUT,114)
 114  FORMAT(1X,T21,'******************************')
C
      READ(KIN,116) CTYPE
 116  FORMAT(A4)
      READ(KIN,*) NUMBT,LMAX,PMAX,VMIN,VMAX
      READ(KIN,*) TCI,TSAT,QSTM,STMLD,PSAT,PD
      READ(KIN,*) TIS,TOTS,TSHL,TTSP,THW,THDR,ITSD
      READ(KIN,*) OTSD,SHLD,TSPD,HWD,HDRD,TBD,ETUBE,TCOVER,COVERD
C
      DO 15 KT= 1, NUMBT
      READ(KIN,*) (TDATA(KT,J),J=1,7)
      READ(KIN,*) (TDATA(KT,J),J=8,12)
  15  CONTINUE
C
      IF(CTYPE .EQ. ITYPE) THEN
      INDEX1= 1
      WRITE(KOUT,118)
 118  FORMAT(1X,'. . CONDENSER TYPE: SUBMARINE DESIGN')
      ELSE
      INDEX1= 2
      WRITE(KOUT,120)
 120  FORMAT(1X,'. . CONDENSER TYPE: SURFACE SHIP DESIGN')
      END IF
C
      WRITE(KSCR,122)
 122  FORMAT(1X,'ENTER "1" FOR SINGLE PASS, OR "2" FOR DOUBLE PASS CON
     1DENSER')
      READ(KSCR,*) NPASS
      IF(NPASS .LE. 1) THEN
```

```
      WRITE(KOUT,124)
 124  FORMAT(1X,'. . SINGLE PASS CONDENSER . .')
      ELSE
      END IF
C
      IF(NPASS .GT. 1) THEN
      WRITE(KSCR,126)
 126  FORMAT(1X,'TWO BASIC CONFIGURATIONS ARE AVAILABLE FOR "DOUBLE PA
     1SS CONDENSERS: '/T5,'(1) CONVENTIONAL RETURN HEADER DESIGN, AND'/T
     15,'(2) "U-TUBE" CONSTRUCTION'/T2,'ENTER 1 OR 2 FOR CONFIGURATION S
     1ELECTION')
      READ(KSCR,*) CONFIG
      ELSE
      END IF
      IF(CONFIG .GT. 2) THEN
      GO TO 900
      ELSE IF(CONFIG .LE. 1) THEN
      WRITE(KOUT,128)
 128  FORMAT(1X,'. . DOUBLE PASS CONDENSER WITH CONVENTIONAL RETURN HE
     1ADER DESIGN.')
      ELSE
      WRITE(KOUT,130)
 130  FORMAT(1X,'. . DOUBLE PASS CONDENSER WITH "U-TUBE" CONSTRUCTION.')
      END IF
 900  CONTINUE
C
      WRITE(KSCR,132)
 132  FORMAT(1X,'ENTER: (1) COOLANT FLOW RATE (GPM)')
      READ(KSCR,*) GPM
C
      WRITE(KSCR,134)
 134  FORMAT(1X,'DO YOU WANT TO INCLUDE IN THIS ANALYSIS THE EFFECTS O
     1F CONDENSATION IN THE'/T2,' PRESENCE OF NON-CONDENSABLE GASES?'/T6
     2,'YES OR NO')
      READ(KSCR,136) GAS
 136  FORMAT(A4)
```

```
      IF((GAS .EQ. NO) .OR. (GAS .EQ. NOE)) THEN
      NON= 2
      ELSE
      NON= 1
      END IF
C
      WRITE(KSCR,138)
 138  FORMAT(1X,'DO YOU WANT TO SPECIFY MAXIMUM MAIN STEAM LANE ENTRAN
     1CE VELOCITY (FT/SEC)?'/T6,'YES OR NO')
      READ(KSCR,140) STM
 140  FORMAT(A4)
      IF((STM .EQ. NO) .OR. (STM .EQ. NOE)) THEN
      INDEX2= 2
      WRITE(KSCR,142)
 142  FORMAT(1X,'PROGRAM WILL SELECT RECOMMENDED MAXIMUM MAIN STEAM LA
     1NE ENTRANCE VELOCITY USING CONDENSER OPERATING PRESSURE AS SELECTI
     1ON CRITERIA.'//)
      ELSE
      INDEX2= 1
      WRITE(KSCR,146)
 146  FORMAT(1X,'ENTER MAXIMUM MAIN STEAM LANE ENTRANCE VELOCITY (FT/S
     1EC)')
      READ(KSCR,*) STMVEL
      END IF
C
      IF(INDEX2 .GT. 1) GO TO 911
      GO TO 912
 911  CONTINUE
      IF(PSAT .LT. 1.) THEN
      STMVEL= 500.
      ELSE IF(PSAT .GT. 5.) THEN
      STMVEL= 200.
      ELSE IF(PSAT .LE. 3.) THEN
      VEL= 600. - 100. * PSAT
      STMVEL= ANINT(VEL)
      ELSE IF(PSAT .GT. 3.) THEN
```

```
        VEL= 450. - 50. * PSAT
        STMVEL= ANINT(VEL)
        ELSE
        END IF
C STMVEL= MAXIMUM MAIN STEAM LANE ENTRANCE VELOCITY (FT/SEC)
 912    CONTINUE
C
        WRITE(KSCR,148)
 148    FORMAT(1X,'DO YOU WANT TO SPECIFY CONDENSATE DRAINAGE PLATE SPAC
     1ING (FT.)?'/T6,'YES OF NO')
        READ(KSCR,150) DRAIN
 150    FORMAT(A4)
        IF((DRAIN .EQ. NO) .OR. (DRAIN .EQ. NOE)) THEN
        INDEX3= 2
        WRITE(KSCR,152)
 152    FORMAT(1X,'PROGRAM WILL SELECT RECOMMENDED CONDENSATE DRAINAGE P
     1LATE SPACING.'//)
        ELSE
        INDEX3= 1
        WRITE (KSCR,154)
 154    FORMAT(1X,'ENTER CONDENSATE DRAINAGE PLATE SPACING (FT.)')
        READ(KSCR,*) DRNPLT
        END IF
C
C * * * * FUNCTIONS FOR THERMAL & MECHANICAL PROPERTIES * * * *
C
        HFG=    1069.185 - 26.72568 * PSAT + 6.659357 * PSAT**2 -
     1          .9288262 * PSAT**3 + .05023536 * PSAT**4
        DENSC= PSAT**2 * .008829 - PSAT * .21843 + 62.347
        KSAT=  (PSAT**2 * (-.05904) + PSAT * 3.0435 + 356.6) * .001
        MU=    5.045709 - .0463856 * TSAT + .1242855E-03 * TSAT**2
        SIGMA= .0057445 - .7136956E-05 * TSAT - .64168E-08 * TSAT**2
        SPVOL= 1207.892 - 756.2292 * PSAT + 209.3 * PSAT**2 -
     1         25.8823 * PSAT**3 + 1.16875 * PSAT**4
C HFG=   LATENT HEAT OF CONDENSATION (BTU/LBM)
C DENSC= DENSITY OF CONDENSATE (LBM/FT**3)
```

```
C KSAT=  CONDUCTIVITY OF CONDENSATE (BTU/HR-FT-DEG.F)
C MU=    ABSOLUTE VISCOSITY OF CONDENSATE (LBM/HR-FT)
C SIGMA= SURFACE TENSION OF CONDENSATE (LBF/FT)
C SPVOL= SPECIFIC VOLUME OF STEAM AT SAT. CONDITIONS (FT**3/LBM)
C
        MF= 1.025
C
        DO 20 NLOOP= 1, 25
C
        DO 25 KT= 1, NUMBT
C
        NTYPE=  TDATA(KT,1)
        DI=     TDATA(KT,2)
        DW=     TDATA(KT,3)
        AE=     TDATA(KT,4)
        AEPE=   TDATA(KT,5)
        EI=     TDATA(KT,6)
        PI=     TDATA(KT,7)
        FLUTE=  TDATA(KT,8)
        MI=     TDATA(KT,9)
        RI=     TDATA(KT,10)
        KW=     TDATA(KT,11)
        RSCALE= TDATA(KT,12)
        WF= 17875.5 / MU * DENSC**2 * AE**3 * AE/AEPE * EXP(-3.33 *AEPE)
C WF= FLOODING CONDENSATE FLOW RATE PER FLUTE (LB/HR)
C
        FE= 3.661218 + 6.4526 * AEPE - 15.265 * AEPE**2 + 16.14543
     1      * AEPE**3 - 6.56116 * AEPE**4
        TDATA(KT,13)= FE
C FE= OPERAND FOR CONDENSING HEAT TRANSFER COEFFICIENT
C
        IF(AE .LE. .015) THEN
        SPEC1= 3.54272 - 522.1174 * AE + .1928462E+05 * AE**2
        ELSE IF (AE .LE. .025) THEN
        SPEC1= .3110063 - 25.2006 * AE + 520.0139 * AE **2
        ELSE
```

```
        SPEC1= .0303163 - 1.345982 * AE + 14.93315 * AE**2
        END IF
C
        NOTE= 0
        DO 30 LOOP1= 1, 250
        IF(NOTE .GT. 0) GO TO 922
        IF(LOOP1 .EQ. 1) THEN
        VM= VMIN / 1.5
        ELSE IF(LOOP1 .EQ. 2) THEN
        VM= VMAX * 1.5
        VDIFF= VMAX * 1.5 - VMIN / 1.5
        ELSE IF(LOOP1 .EQ. 3) THEN
        VM= (VMAX * 1.5 + VMIN / 1.5) / 2.
        ELSE
        END IF
C
C * * * * CALCULATION OF BASIC CONDENSER DATA * * * *
C
 922    QT= STMLD * QSTM * HFG
        AW= GPM * .0022283 / VM
        WC= AW * VM * 230400.
C QT= CONDENSER HEAT LOAD (BTU/HR)
C AW= TOTAL CROSS-SECTIONAL AREA REQUIRED FOR COOLANT FLOW (FT**2)
C WC= COOLING WATER FLOW RATE (LB/HR)
C
        TOUT=  TCI + QT / (.95 * WC)
        ARG1=  DI / 24.
        NT=    AW / (PIE * ARG1**2)
        NT=    ANINT(NT)
        TNT=   NT * NPASS
        DN=    DW + AE * 2.
        OD=    DW + AE * 4.
        DTEMP= (TSAT - TCI) / (TSAT - TOUT)
        ODTLM= (TOUT - TCI) / LOG(DTEMP)
        OTCS=   (TOUT + TCI) / 2.
        CNU= .122181 - .148162E-02 * OTCS + .5516445E-05 * OTCS**2
```

```
C TOUT=  OUTLET COOLANT TEMPERATURE (DEG.F)
C NT=    NUMBER OF TUBES PER CONDENSER PASS
C TNT=   TOTAL NUMBER OF TUBES
C DN=    NOMINAL TUBE DIAMETER (IN.)
C OD=    OUTSIDE TUBE DIAMETER (IN.)
C ODTLM= CONDENSER LOG MEAN TEMPERATURE DIFFERENCE (DEG.-F)
C
         CALL GEOM(OD,TNT)
C
C * * * * DETERMINATION OF CONDENSATE DRAINAGE PLATE SPACING * * * *
C
         MACH=  STMVEL / SQRT(3672.76 * (TSAT + 459.67))
         CDRAG= .75 * MACH + .975
         PITCH= PD * OD
C MACH=  MACH NUMBER OF STEAM INLET FLOW TO CONDENSER
C CDRAG= DRAG COEFFICIENT FOR STEAM INLET FLOW OVER CONDENSER TUBES
C PD=    TUBE PITCH-TO-DIAMETER RATIO
C PITCH= TUBE PITCH (IN.)
C
         IF(INDEX3 .LE. 1) GO TO 934
         DRNPLT= 2.5
C
         DO 35 N1= 1, 60
C
         WDRAG= CDRAG * OD * DRNPLT * STMVEL**2 / (24. * SPVOL)
C WDRAG= DRAG FORCE LOADING ON TUBES DUE TO STEAM VELOCITY(LBF.)
         DELT= 5.
C "DRNPLT=2.5" & "DELT=5." ARE BOTH ASSUMED VALUES TO INITIATE THE
C FOLLOWING ITERATIVE CALCULATIONS.
C
         IF(N1 .EQ. 1) THEN
         TMIN1= TCI + .15
         T2=   TMIN1
         ELSE IF(N1.EQ.2)THEN
         TMAX1=  TCI + 20.
         T2=    TMAX1
```

```
          TDIFF1= TMAX1- TMIN1
          ELSE IF(N1 .EQ. 3) THEN
          T2= TCI + TDIFF1 / 2.
          ELSE
          END IF
C
          DTS= T2 - TCI
          TCS= TCI + DTS / 2.
          ARG7= (TSAT - TCI) / (TSAT - T2)
          DTLM= DTS / LOG(ARG7)
C DTLM= LOG MEAN TEMPERATURE DIFFERENCE (DEG.F)
C THIS ANALYSIS EXAMINES A SAMPLE TUBE SECTION OF LENGTH= DRNPLT TO
C DETERMINE RECOMMENDED VALUES FOR CONDENSATE DRAINAGE PLATE SPACING.
C THIS TUBE SECTION IS LOCATED IN THE COOLING WATER INLET SECTION OF THE
C CONDENSER, WHERE THE CONDENSATION RATE/PER UNIT TUBE LENGTH WILL BE AT
C A MAXIMUM.
C DRNPLT= CONDENSATE DRAINAGE PLATE SPACING (FT.)
C DELT=    SAT. TEMP. - TUBE WALL TEMP. (DEG.F)
C T2=      TUBE SECTION C.W. OUTLET TEMP. (DEG.F)
C DTS=     C.W. TEMP. RISE ACROSS TUBE SECTION (DEG.F)
C TCS=     AVGERAGE C.W. TEMP. FOR EACH SECTION (DEG.F)
C DTLM=    LOG MEAN TEMPERATURE DIFFERENCE (DEG.F)
C
          IF(NTYPE .LE. 1) THEN
          HW= .023 * KSW(TCS) / DI * 12. * RE(VM)**.8 * PRSW(TCS)**.4
C HW= HEAT TRANSFER COEFFICIENT COOLANT SIDE FOR SMOOTH INTERNAL TUBE
C     (BTU/HR-FT**2-DEG.F)
          ELSE
          ARG2= 7. / RE(VM)
          GAMMA= -1. * (LOG(2. * EI/DI) * 2.5 + 3.75)
          ARG3=  EI / PI
          ARG4=  EI / DI * RE(VM) * FF(ARG2)
          HW=    12. * KSW(TCS) * RE(VM) * PRSW(TCS) * ST(ARG3,ARG4) / DI
C HW= HEAT TRANSFER COEFFICIENT COOLANT SIDE FOR DOUBLY ENHANCED TUBES
C     (BTU/HR-FT**2-DEG.F)
          END IF
```

```
C
        DO 40 N3= 1, 10
C
        ARG5= QSTM * HFG * WF / (DRNPLT * DELT)
        ARG6= KSAT**3 * DENSC*SIGMA*QSTM*HFG * 4.17*10.**8/(MU*DELT)
        ARG8= AE / 12.
        HCOND= 7.2324 * ARG5**.0774 * ARG8**.2307 * FE**.9226 /(AE/AEPE)
     1          * ARG6**.2307
C HCOND= HEAT TRANSFER COEFFICIENT CONDENSATE (BTU/HR-FT**2-DEG.F)
C
        US= 1. / (DN/(DI*HW)+RSCALE + 1./HCOND + LOG(DW/DI *(7.985 *
     1    (AE/12.)**.7688 * AEPE**-.0242 + 1)) * DN /(24. * KW))
C US= OVERALL HEAT TRANSFER COEFFICIENT (BTU/HR-FT**2-DEG.F)
C
        QS=    .95 * WC * DTS
        DELT= QS * 12. / (NT * PIE * DN * DRNPLT * HCOND)
C QS= HEAT LOAD FOR TUBE SECTION (BTU/HR)
  40    CONTINUE
C
        L1= QS * 12. / (NT * PIE * DN * US * DTLM)
        WCWF= QS / (NT * QSTM* HFG * WF * FLUTE)
        DEFL= .26526*WDRAG*DRNPLT**3 / (ETUBE*(OD**4 - DI**4))
C DEFL= MAXIMUM TUBE DEFLECTION (IN.)
C
        IF(N1 .EQ.1) THEN
        LMIN1= L1
        WMIN1= WCWF
        GO TO 933
        ELSE IF(N1 .EQ. 2) THEN
        LMAX1=   L1
        WMAX1=   WCWF
        DIFFL1=  LMAX1 - LMIN1
        WDIFF1=  WMAX1 - WMIN1
        GO TO 933
        ELSE
        END IF
```

```
      IF(ABS(DRNPLT - L1) .GT. .005) THEN
      MULT2= (DRNPLT - L1) / DIFFL1
      STEP1= MULT2 * TDIFF1
      T2=    T2 + STEP1
      GO TO 933
      ELSE IF(ABS(SPEC1 - WCWF) .GT. .001) THEN
      MULT3= (SPEC1 - WCWF) / WDIFF1
      STEP2= MULT3 * DIFFL1
      DRNPLT= DRNPLT + STEP2
      GO TO 933
      ELSE IF(DEFL .GT. PITCH/2.) THEN
      DRNPLT= DRNPLT - .05
      ELSE
      END IF
      IF((ABS(DRNPLT-L1) .LT. .005) .AND. (ABS(SPEC1-WCWF) .LT. .001)
     1    .AND. (DEFL .LT. PITCH/2.)) GO TO 934
  933 CONTINUE
   35 CONTINUE
  934 CONTINUE
      DEFL= 1375. * WDRAG * DRNPLT**3 / (ETUBE * (OD**4 - DI**4))
      IF(DEFL .GT. PITCH/2.) THEN
      WRITE(KOUT,156) DEFL
      WRITE(KSCR,156) DEFL
  156 FORMAT(1X,'* * PROGRAM WARNING NR-3: SPECIFIED CONDENSATE DRAINA
     1GE PLATE SPACING IS BEYOND TUBE SUPPORT LIMIT * *'/T2,'. . MAXIMUM
     1 TUBE DEFLECTION= ',F6.3,' (IN.)')
      ELSE
      END IF
C * * * * CALCULATION OF EFFECTIVE TUBE LENGTH * * * *
C
      SUML=  0.0
      SUMQ=  0.0
      AVGHW= 0.0
      AVGHC= 0.0
      AVGU=  0.0
      T1= TCI
```

```
C
      DO 45 N4= 1, 100
C
      DELT= 5.
C
      DO 50 N5= 1, 25
C
      IF(N5 .EQ. 1) THEN
      TMIN= T1 + .15
      T2=   TMIN
      ELSE IF(N5 .EQ. 2) THEN
      TMAX=  T1 + 20.
      T2=    TMAX
      TDIFF= TMAX - TMIN
      ELSE IF(N5 .EQ. 3) THEN
      T2=    T1 + TDIFF / 2.
      ELSE
      END IF
C
      DTS=  T2 - T1
      TCS=  T1 + DTS / 2.
      ARG7= (TSAT - T1) / (TSAT - T2)
      DTLM= DTS / LOG(ARG7)
C
      IF(NTYPE .LE. 1) THEN
      HW= .023 * KSW(TCS)/DI * 12. * RE(VM)**.8 * PRSW(TCS)**.4
      ELSE
      ARG2= 7. / RE(VM)
      GAMMA= -1. * (LOG(2. * EI/DI) * 2.5 + 3.75)
      ARG3= EI / PI
      ARG4= EI / DI * RE(VM) * FF(ARG2)
      HW= 12. * KSW(TCS) * RE(VM) * PRSW(TCS) * ST(ARG3,ARG4) / DI
      END IF
C
      DO 55 N6= 1, 10
C
```

```
      ARG5= QSTM * HFG * WF / (DRNPLT * DELT)
      ARG6= KSAT**3 * DENSC * SIGMA * QSTM * HFG*4.17*10.**8/(MU*DELT)
      ARG8= AE / 12.
      HCOND= 7.2324 * ARG5**.0774 * ARG8**.2307 * FE**.9226 /(AE/AEPE)
     1         * ARG6**.2307
C
      US= 1. / (DN/(DI*HW)+RSCALE + 1./HCOND +LOG(DW/DI * (7.985 *
     1   (AE/12.)**.7688 * AEPE**-.0242 + 1)) * DN / (24. * KW))
      QS= .95 * WC * DTS
      DELT= QS * 12. / (NT * PIE * DN * DRNPLT * HCOND)
   55 CONTINUE
C
      L1= QS * 12. / (NT * PIE * DN * US * DTLM)
C
      IF(N5 .EQ. 1) THEN
      LMIN2= L1
      GO TO 944
      ELSE IF(N5 .EQ. 2) THEN
      LMAX2=  L1
      DIFFL2= LMAX2 - LMIN2
      GO TO 944
      ELSE
      END IF
      IF(ABS(DRNPLT - L1) .GT. .005) THEN
      MULT4= (DRNPLT - L1) / DIFFL2
      STEP4= MULT4 * TDIFF
      T2= T2 + STEP4
      ELSE
      END IF
      IF(ABS(DRNPLT - L1) .LT. .005) GO TO 945
C
  944 CONTINUE
   50 CONTINUE
C
  945 WCWF= QS / (NT * QSTM * HFG * WF * FLUTE)
      IF(WCWF .GT. SPEC1 + .005) THEN
```

```
        WRITE(KOUT,158) N4
        WRITE(KSCR,158) N4
 158    FORMAT(1X,'* * PROGRAM WARNING NR-2: "WCWF" BEYOND SPECIFICATION
     1 LIMITS - SECTION (',I3,') * * ')
        ELSE
        END IF
C
        DATA(N4,1)= QS
        DATA(N4,2)= HW
        DATA(N4,3)= HCOND
        DATA(N4,4)= US
        DATA(N4,5)= DRNPLT
C * * CALCULATION OF FINAL SECTION LENGTH * *
        SUML=  DATA(N4,5) + SUML
        SUMQ=  DATA(N4,1) + SUMQ
        AVGHW= DATA(N4,2) + AVGHW
        AVGHC= DATA(N4,3) + AVGHC
        AVGU=  DATA(N4,4) + AVGU
        DELT= 5.
        QFS= QT - SUMQ
        TCS= (TOUT + T2)/2.
C
        IF(NTYPE .LE. 1) THEN
        HW= .023 * KSW(TCS)/DI * 12. * RE(VM)**.8 * PRSW(TCS)**.4
        ELSE
        ARG2= 7./ RE(VM)
        ARG4= EI / DI * RE(VM) * FF(ARG2)
        HW= 12. * KSW(TCS) * RE(VM) * PRSW(TCS) * ST(ARG3,ARG4) / DI
        END IF
C
        TL= 2.0
        DO 60 N7= 1, 10
        ARG5=  QSTM * HFG * WF / (TL * DELT)
        ARG6= KSAT**3 * DENSC * SIGMA * QSTM * HFG*4.17*10.**8/(MU*DELT)
        HCOND= 7.2324 * ARG5**.0774 * ARG8**.2307 * FE**.9226 /(AE/AEPE)
     1         * ARG6**.2307
```

```
      US= 1. /(DN/(DI*HW)+RSCALE + 1./HCOND + LOG(DW/DI * (7.985 *
     1    (AE/12.)**.7688 * AEPE**-.0242 + 1)) * DN/ (24. * KW))
        ARG7= (TSAT - T2) / (TSAT - TOUT)
        DTLM= (TOUT - T2) / LOG(ARG7)
        QP=   QFS * 12. / (PIE * DN * NT)
        TL=   QP / (US * DTLM)
        DELT= QP / (TL * HCOND)
  60    CONTINUE
C
        IF(TL .LE. DRNPLT) THEN
        NS = N4 + 1
        SUML= SUML + TL
        AVGHW= AVGHW / NS
        AVGHC= AVGHC / NS
        AVGU=  AVGU  / NS
        GO TO 950
        ELSE
        T1= T2
        END IF
  45    CONTINUE
 950    HWAVG(KT)= AVGHW
        HCAVG(KT)= AVGHC
        UAVG(KT)=  AVGU
C
C * * CONDENSER WEIGHT & VOLUME CALCULATIONS * *
C
        EFF(KT)= SUML / NPASS
        TLT(KT)= SUML / NPASS * MF
C
        IF(INDEX1 .LE. 1) THEN
        LTOT= (9.125 + TIS) / 6. + TLT(KT)
        ELSE
        LTOT= TIS / 6. + TLT(KT)
        END IF
C
        LANE= 1.22E-04 * STMLD * SPVOL / (TLT(KT) * STMVEL)
```

```
C TLT = TOTAL EFFECTIVE TUBE LENGTH (FT.),(TLT=TOTL)
C LTOT= TOTAL TUBE LENGTH (FT.)
C LANE= STEAM LANE BREADTH (FT.)
C
        ATB=    PIE * TOTRAD**2
        DTB=    SQRT(4. * ATB / PIE)
        ARG10= (DTB + 2. * LANE) / 2.
        ATS=    PIE * ARG10**2
        DTS=    DTB + 2 * LANE
        VTB=    TLT(KT) * ATB
C ATB= AREA OF TUBE BUNDLE (FT**2)
C DTB= DIAMETER OF TUBE BUNDLE (FT.)
C ATS= AREA OF TUBE SHEET (FT**2)
C DTS= DIAMETER OF TUBE SHEET (FT.)
C VTB= VOLUME OF TUBE BUNDLE (FT**3)
C
        DSHL=    DTS + TSHL / 6.
        DPTHDR= DTS / 2.
        VHW=     STMLD/3690.-.0417*ATB*NS/NPASS-.2618*TLT(KT)
        DHW=     DSHL + 1
        LHW=     VHW / (.7854 * DHW**2) + DPTHDR + .5
        CONDH=   LHW + DPTHDR + LTOT
C DSHL=    OUTER DIAMETER OF SHELL (FT.)
C DPTHDR= HEADER DEPTH (FT.)
C LHW=     LENGTH OF HOTWELL (FT.)
C VHW=     VOLUME OF HOTWELL (FT**3)
C CONDH=   TOTAL CONDENSER HEIGHT (FT.)
C
        CALL DELTAP(DTB,CNU,LTOT,CONFIG,NPASS,DI,VM,GPM,
     1              TOTHL,SYSL,PMPWR)
        IF(LOOP1 .EQ. 1) THEN
        LMIN= CONDH
        PMIN= PMPWR
        GO TO 955
        ELSE IF(LOOP1 .EQ. 2) THEN
        LPLUS= CONDH
```

```
          PPLUS= PMPWR
          DIFFL= LPLUS - LMIN
          DIFFP= PPLUS - PMIN
          GO TO 955
          ELSE
          END IF
C
          IF(NOTE .GT. 0) GO TO 964
          IF(ABS(LMAX - CONDH) .GT. .005) THEN
          MULT1= (LMAX - CONDH) / DIFFL
          CHANGE= MULT1 * VDIFF
          VM= VM + CHANGE
          GO TO 955
          ELSE
          END IF
 964      IF(VM .LT. VMIN) THEN
          GO TO 956
          ELSE IF(VM .GT. VMAX) THEN
          VM= .99 * VM
          NOTE= 1
          ELSE IF(PMPWR .GT. PMAX) THEN
          MULTP= (PMAX - PMPWR) / DIFFP
          PSTEP= MULTP * VDIFF
          VM=     VM + PSTEP
          NOTE= 1
          ELSE
          END IF
          IF(NOTE .GT. 0) THEN
          GO TO 965
          ELSE
          GO TO 966
          END IF
 965      IF(ABS(PMAX - PMPWR) .LE. .005) GO TO 956
          GO TO 955
 966      IF((ABS(LMAX - CONDH) .LE. .005) .AND. (PMPWR .LE. PMAX))
     1        GO TO 956
```

```
 955     CONTINUE
         IF(LMIN .GT. LMAX) THEN
         GO TO 956
         ELSE
         END IF
   30    CONTINUE
C
 956     CONTINUE
         XLTOT(KT)= CONDH
         XVM(KT)=   VM
         NOTE= 0
         WRITE(KSCR,160) KT
 160     FORMAT(T2,'* * TUBE NO.(',I3,') TESTED * *')
   25    CONTINUE
C
         XCOMP= 0.0
C
         DO 65 KT1= 1, NUMBT
         IF(XVM(KT1) .GT. XCOMP) THEN
         XCOMP= XVM(KT1)
         OPT= KT1
         ELSE
         XCOMP= XCOMP
         END IF
  65     CONTINUE
C
         AVGHW=   HWAVG(OPT)
         AVGHC=   HCAVG(OPT)
         AVGU=    UAVG(OPT)
         EFFL=    EFF(OPT)
         TOTL=    TLT(OPT)
         CONDH=   XLTOT(OPT)
         VM=      XVM(OPT)
         NTYPE=   TDATA(OPT,1)
         DI=      TDATA(OPT,2)
         DW=      TDATA(OPT,3)
```

```
      AE=      TDATA(OPT,4)
      AEPE=    TDATA(OPT,5)
      EI=      TDATA(OPT,6)
      PI=      TDATA(OPT,7)
      FLUTE=   TDATA(OPT,8)
      MI=      TDATA(OPT,9)
      RI=      TDATA(OPT,10)
      KW=      TDATA(OPT,11)
      RSCALE=  TDATA(OPT,12)
      FE=      TDATA(OPT,13)
C
      WRITE(KSCR,162)OPT,VM
 162  FORMAT(/T2,'* * TUBE NO.(',I3,') SELECTED AT V= ',F5.2,
     1       ' (FT/SEC) * *'//)
C
      IF(NON .LE. 1) GO TO 977
      GO TO 978
 977  CALL NONCOND(ODTLM,DN,OD,TNT,NPASS,TOTNC)
      TOTL= TOTNC
      WRITE(KSCR,164)
 164  FORMAT(T2,'* * NON-CONDENSABLE GAS ANALYSIS * *'/)
C
 978  IF(INDEX1 .LE. 1) THEN
      LTOT= (9.125 + TIS) / 6. + TOTL
      ELSE
      LTOT= TIS / 6. + TOTL
      END IF
      LANE=  1.22E-04 * STMLD * SPVOL / (TOTL * STMVEL)
      ATB=   PIE * TOTRAD**2
      DTB=   SQRT(4. * ATB / PIE)
      ARG10= (DTB + 2. * LANE) / 2.
      ATS=   PIE * ARG10**2
      DTS=   DTB + 2 * LANE
      VTB=   TOTL * ATB
C
      ARG12= DTS - 2 * LANE + OD / 6.
```

```
      WP1=    .785398 * (ARG12**2 - OD**2 * TNT * .00694)
      WP2=    .785398 * (DTS**2 - OD**2 *TNT* .00694)
      WOTS=   OTSD * TOTS / 6. * WP2
      WITS=   ITSD * TIS / 6. * WP2
      WTSP=   NS / NPASS * TSPD * TTSP/12. * WP1
      WTB=    .005454 * (DN**2 - DI**2) * LTOT * TBD * TNT
      DSHL=   DTS + TSHL / 6.
      DPTHDR= DTS / 2.
      WSHL=   .785398 * SHLD * LTOT * (DSHL**2 -DTS**2)
C WOTS= WEIGHT OF OUTER TUBE SHEETS (LB.)
C WITS= WEIGHT OF INNER TUBE SHEETS (LB.)
C WTSP= WEIGHT OF TUBE SUPPORT PLATES (LB.)
C WTB=  WEIGHT OF TUBE BUNDLE (LB.)
C WSHL= WEIGHT OF SHELL (LB.)
C DSHL= OUTER DIAMETER OF SHELL (FT.)
C DPTHDR= HEADER DEPTH (FT.)
C
      WEXP= .39167 * DTB * TSHL * SHLD
      WHDR= .4083 * DTB**2 * THDR * HDRD
      VHW=  STMLD / 3690. - .0417 * ATB * NS/NPASS - .2618 * TOTL
      DHW=  DSHL + 1.
      LHW=  VHW / (.7854 * DHW**2) + DPTHDR + .5
      WHW=  .2618 * (DHW**2 / 4. + DHW * LHW) * THW * HWD
C LHW=  LENGTH OF HOTWELL (FT.)
C VHW=  VOLUME OF HOTWELL (FT.)
C WEXP= WEIGHT OF EXPANSION JOINT (LB.)
C WHDR= WEIGHT OF HEADER (LB.)
C WHW=  WEIGHT OF HOTWELL (LB.)
C
      IF(CONFIG .GT. 2) THEN
      GO TO 988
      ELSE
      END IF
      IF(CONFIG .GT. 1) THEN
      WCOVER= .20415 * DTB**2 * TCOVER * COVERD
      WHDR= WHDR/2. + WCOVER
```

```
      WUTUBE= .005454 * (DN**2 - DI**2) * DTB * TBD * TNT
      WTB=  .005454 * (DN**2 - DI**2) * LTOT * TBD * TNT + WUTUBE
      ELSE
      END IF
 988  CONTINUE
      WT=    WOTS + WITS + WTSP + WTB + WSHL + WEXP + WHDR + WHW
      WMISC= WT * .25
      WDRY=  WT * 1.25
      WLIQ=  .005454 * DI**2 *NT*LTOT + 33.51 * DSHL**3 +STMLD/226935.
      WET=   WDRY + WLIQ
      TONS=  WET / 2240.
C WMISC= WEIGHT OF MISCELLANEOUS COMPONENTS (LB.)
C WDRY=  TOTAL DRY WEIGHT OF CONDENSER (LB.)
C WLIQ=  TOTAL LIQUID WEIGHT (LB.)
C WET=   TOTAL WET WEIGHT OF CONDENSER (TONS)
C
      IF(NON .LT. 2) THEN
      CONDHC=  LHW + DPTHDR + LTOT
      CONDL=   CONDHC
      ELSE
      CONDH=   LHW + DPTHDR + LTOT
      CONDL=   CONDH
      END IF
C
      BOXVOL=  DSHL**2 * (LTOT + DPTHDR) + DHW**2 * LHW
C CONDH=  CONDENSER HEIGHT (FT.)
C BOXVOL= CONDENSER ENCLOSED BOX VOLUME (FT**3)
C
      ARG14= (LHW + LTOT/2.) * (WOTS + WITS + WTSP + WSHL + WEXP)
     1       + (LHW/2.) * WHW + CONDL/2. * WMISC
      IF(CONFIG .GT. 1) THEN
      DRYCG= (ARG14 +(LHW - DPTHDR/2.)*WDHDR/2. + (LHW+LTOT+DPTHDR/2.)
     1 * (WCOVER + WUTUBE) + (LHW + LTOT/2.) * WTB) / WDRY
      ELSE
      DRYCG= (ARG14 + (LHW + LTOT/2.) * (WTB + WHDR)) / WDRY
      END IF
```

```
      ARG15= STMLD / 3690. * 61.5
      ARG16= WLIQ - ARG15
      WETCG= (DRYCG * WDRY + ARG16 * (LHW + LTOT/2.) + ARG15 * VHW /
     1       25.1328) / WET
C DRYCG=HEIGHT OF CENTER OF GRAVITY ABOVE CONDENSER BOTTOM FOR CONDENSER
C       AT DRY WEIGHT (FT.)
C WETCG=HEIGHT OF CENTER OF GRAVITY ABOVE CONDENSER BOTTOM FOR CONDENSER
C       AT WET WEIGHT (FT.)
C
      CALL DELTAP(DTB,CNU,LTOT,CONFIG,NPASS,DI,VM,GPM,
     1            TOTHL,SYSL,PMPWR)
C
      A= N
      OUTPUT(N,1)= A
      OUTPUT(N,2)= DSHL
      OUTPUT(N,3)= CONDL
      OUTPUT(N,4)= BOXVOL
      OUTPUT(N,5)= WET
      OUTPUT(N,6)= PMPWR
C
C * * * * CALCULATION RESULTS - OUTPUT(DATA FILE) * * * *
C
      IF(NTYPE .LE. 1) THEN
      WRITE(KOUT,166)
 166  FORMAT(T2,'. . SMOOTH INTERNAL TUBES . .')
      ELSE
      WRITE(KOUT,168)
 168  FORMAT(T2,'. . DOUBLY ENHANCED TUBES . .')
      END IF
C
      WRITE(KOUT,170) OPT
 170  FORMAT(T2,'. . TUBE NUMBER(',I3,') SELECTED . .')
      IF(NON .LE. 1) THEN
      WRITE(KOUT,172)
 172  FORMAT(T2,'. . NON-CONDENSABLE GAS ANALYSIS . .')
      ELSE
```

```
      END IF
      WRITE(KOUT,174)
174   FORMAT(T2,T27,'**** DATA INPUT ****')
      WRITE(KOUT,176) GPM,VM
176   FORMAT(T2,'. . FLOW RATE: ',F6.0,' (GPM)  . . FLOW VELOCITY: ',
     1        F5.2,' (FT/SEC) . .')
      WRITE(KOUT,178) DI,DW,AE,AEPE,NTYPE,EI,PI,FE,MI,RI,KW,TCI,PSAT,
     1                QSTM,TSAT,STMLD,RSCALE
178   FORMAT(T2,'DI = ',F5.3,T16,'DW = ',F5.3,T30,'AE = ',F5.3,T44,'AE
     1PE = ',F6.4,T61,'NTYPE = ',I2/T2,'EI = ',F6.4,T16,'PI = ',F6.4,T30
     1,'FE = ',F5.2,T44,'MI = ',F5.3,T61,'RI = ',F6.4/T2,'KW = ',F6.2,T1
     16,'TCI = ',F6.2,T30,'PSAT = ',F4.2,T44,'QSTM = ',F4.2,T61,'TSAT =
     1',F6.2/T2,'STMLD = ',F9.1,T30,'RSCALE = ',F7.5/)
      WRITE (KOUT,180)
180   FORMAT(T17,'**** DATA OUTPUT - CONDENSER SIZING ROUTINE ****')
      WRITE(KOUT,182) STMVEL
182   FORMAT(T2,'. . MAXIMUM MAIN STEAM LANE ENTRANCE VELOCITY: ',F4.0
     1,' (FT/SEC)')
      WRITE(KOUT,184) DRNPLT
184   FORMAT(T2,'. . CONDENSATE DRAINAGE PLATE SPACING: ',F5.2,
     1          ' (FT.)'/)
      WRITE(KOUT,186) QT,TOUT,TNT
186   FORMAT(1X,'. . CONDENSER HEAT LOAD:   ',E12.5,' (BTU/HR)'/T2,'.
     1. OUTLET COOLANT TEMP:    ',F6.2,' (DEG.F)'/T2,'. . TOTAL NUMBER O
     1F TUBES: ',F7.1/)
      WRITE(KOUT,188) TOTL,AVGHW,AVGHC,AVGU
188   FORMAT(1X,'. . EFFECTIVE TUBE LENGTH:',T39,F5.2,' (FT.)'/T2,'. .
     1 AVG. HEAT TRNFR. COEFF. C.W.:',T38,F6.0,' (BTU/HR-FT**2-DEG.F)'/T
     12,'. . AVG. HEAT TRNFR. COEFF. COND.:',T38,F6.0,' (BTU/HR-FT**2-DE
     1G.F)'/T2,'. . AVG. OVERALL HEAT TRNFR. COEFF.:',T38,F6.0,' (BTU/HR
     1-FT**2-DEG.F)'/)
      WRITE(KOUT,190) LTOT,LANE,DTB,ATB,VTB,DTS,ATS,WEXP,WMISC,LHW,
     1                DPTHDR
190   FORMAT(1X,'LTOT = ',F5.2,T19,'LANE = ',F5.2,T37,'DTB = ',F6.2,T5
     13,'ATB = ',F6.2/T2,'VTB = ',F6.2,T19,'DTS = ',F5.2,T37,'ATS = ',F6
     1.2///T12,'**** DATA OUTPUT - WEIGHT & VOLUME CALCULATIONS ****'//T
```

```
     12,'WEXP = ',F7.1,T19,'WMISC = ',F7.1,T37,'LHW = ',F5.2,T53,'DPTHDR
     1 = ',F5.2///T30,'MATERIAL',T47,'THICKNESS',T64,'WEIGHT'/T26,'DENSI
     1TY(LB/FT**3)',T49,'(IN.)',T65,'(LB.)')
        IF(INDEX1 .GT. 1) THEN
        WRITE(KOUT,192) ITSD,TIS,WITS
 192    FORMAT(T2,'TUBE SHEET-----------',T31,F5.1,T49,F6.4,T64,F6.0)
        ELSE
        WRITE(KOUT,194) OTSD,TOTS,WOTS,ITSD,TIS,WITS
 194    FORMAT(T2,'OUTER TUBE SHEET-----',T31,F5.1,T49,F6.4,T64,F6.0/T2,
     1          'INNER TUBE SHEET-----',T31,F5.1,T49,F6.4,T64,F6.0)
        END IF
        WRITE(KOUT,198) TSPD,TTSP,WTSP,TBD,WTB,SHLD,TSHL,WSHL,HWD,THW,
     1                  WHW,HDRD,THDR,WHDR,WDRY
 198    FORMAT(T2,'TUBE SUPPORT PLATE---',T31,F5.1,T49,F6.4,T64,F6.0/T2,
     1       'TUBE BUNDLE----------',T31,F5.1,T49,'*.****',T64,F6.0/T2,
     1           'CONDENSER SHELL------',T31,F5.1,T49,F6.4,T64,F6.0/T2,
     1           'HOTWELL--------------',T31,F5.1,T49,F6.4,T64,F6.0/T2,
     1           'WATERBOX-------------',T31,F5.1,T49,F6.4,T64,F6.0//T2
     1          ,'. . TOTAL CONDENSER DRY WEIGHT=  ',F8.1,' (LB.)')
        WRITE(KOUT,200) WET,TONS,CONDL,DSHL,BOXVOL,DRYCG,WETCG,TOTHL,
     1  SYSL,PMPWR
 200    FORMAT(1X,'. . TOTAL CONDENSER WET WEIGHT=  ',F8.1,' (LB.)'/T35,
     1F6.2,' (TON)'//T2,'. . TOTAL CONDENSER HEIGHT= ',F6.2,' (FT.)'/T2,
     1'. . OUTER SHELL DIAMETER=   ',F6.2,' (FT.)'/T2,'. . ENCLOSED BOX
     1VOLUME=   ',F7.1,' (FT**3)'/T6,'DRYCG = ',F6.2,T24,'WETCG = ',F6.2
     1//T2,'. . CONDENSER FRICTIONAL HEAD LOSS= ',F5.1,' (FT.)'/T2,'. .
     1TOTAL SYSTEM HEAD LOSS=',T38,F5.1,' (FT.)'/T2,'. . TOTAL SYSTEM PU
     1MPING POWER=',T38,F5.1,' (HP)'///)
C
        IF(NON .LE. 1) GO TO 998
        GO TO 999
 998    IF(CONDHC .GT. CONDH) THEN
        MF= MF + .01
        ELSE IF(CONDHC .LT. CONDH) THEN
        MF= MF - .01
        ELSE
```

```
        END IF
        IF((ABS(CONDHC-CONDH) .LE. .005*CONDH) .AND. (CONDHC .LE.
     1      CONDH)) GO TO 999
C
        WRITE(KSCR,202) NLOOP
 202    FORMAT(T2,'* * NON-CONDENSABLE GAS ANALYSIS: ITERATION NR:('
     1          ,I3,') * *')
C
  20    CONTINUE
 999    CONTINUE
  10    CONTINUE
C
        WRITE(KOUT,204)
 204    FORMAT('1',T26,'**** DATA SUMMARY ****'//T4,'CONDENSER',T16,'SHE
     1LL',T27,'TOTAL',T36,'ENCLOSED',T48,'TOTAL WET',T61,'TOTAL SYSTEM'/
     1T5,'NUMBER',T15,'DIAMETER',T26,'HEIGHT',T35,'BOX VOLUME',T49,'WEIG
     1HT',T60,'PUMPING POWER'/T16,'(FT.)',T27,'(FT.)',T36,'(FT**3)',T50,
     1'(LB.)',T64,'(HP)')
C
        DO 70 N8= 1, NCOND
        WRITE(KOUT,206)(OUTPUT(N8,J),J=1,6)
 206    FORMAT(T6,F3.0,T15,F6.2,T26,F6.2,T36,F7.1,T49,F7.0,T63,F5.1)
  70    CONTINUE
        STOP
        END
C
        SUBROUTINE DELTAP(DTB,CNU,LTOT,CONFIG,NPASS,DI,VM,GPM,
     1                    TOTHL,SYSL,PMPWR)
C SUBROUTINE DELTAP CALCULATES THE COOLING WATER SIDE PRESSURE DROPS,
C AND EVALUATES THE CONDENSER AND TOTAL SYSTEM PUMPING POWER
        REAL LTOT
        COMMON STMLD,TSAT,PSAT,STM,TOTRAD,ITOT,LANE,QSTM,PD,EFFL,
     1         TOTL,QT,AVGU,RADIUS(250),NTUBES(250)
        RE= VM * DI * 300. / CNU
        IF(NTYPE .EQ. 2) THEN
        FFT= 8. * (-1. / (2.46 * LOG((7/RE)**MI + RI)))**2
```

```
        ELSE
        ARG1= 1. / (7/RE)**.8 + 1.62E-5 / DI
        A=    (2.457 * LOG(ARG1))**16
        B=    (37530 / RE)**16
        FFT=  8. * ((8. / RE)**12 + 1. / (A + B)**1.5)**.0833
        END IF
        IF(CONFIG .EQ. 2) THEN
        FFT= FFT * (RE *(DI / (12 * DTB))**2)**.05
        PWO= .00237 * VM**2.33
        PTE= .0178  * VM**2
        ELSE IF(NPASS .EQ. 1)THEN
        PWO= .00237 * VM**2.33
        PTE= .0178  * VM**2
        ELSE
        PWO= .00635 * VM**2.1
        PTE= .0356  * VM**2
        END IF
        HT= FFT * LTOT * NPASS * VM**2 / (DI * 5.353)
        PWI= .01422 * VM**2.04
C
        HLP= 79.7 - .00338 * GPM
C HT=  FRICTIONAL HEAD LOSS FOR TUBES (FT.)
C HWB= WATERBOX INLET & OUTLET LOSSES (FT.)
C HE=  TUBE END LOSSES (FT.)
C HLP= SEAWATER CIRCULATING LOOP PIPING LOSSES(FT.)
C
        TOTHL= HT + PWO + PTE + PWI
        SYSL=  TOTHL + HLP
        PMPWR= .000352 * GPM * SYSL
C TOTHL= TOTAL FRICTIONAL HEAD LOSS FOR CONDENSER (FT.)
C SYSL=TOTAL FRICTIONAL HEAD LOSS OF CONDENSER AND SYSTEM PIPING(FT.)
C PMPWR= TOTAL SYSTEM PUMPING POWER (HP)
C
        RETURN
        END
C
```

```
      SUBROUTINE GEOM(OD,TNT)
C SUBROUTINE: RADIUS DETERMINES THE TUBE SHEET GEOMETRY
      DIMENSION RAD(250),NTUB(250)
      REAL NTUBES,NTUB
      COMMON STMLD,TSAT,PSAT,STM,TOTRAD,ITOT,LANE,QSTM,PD,EFFL,
     1       TOTL,QT,AVGU,RADIUS(250),NTUBES(250)
      DATA PIE/3.1415927/,VOIDID/.83333/
C VOIDID= CENTRAL VOID DIAMETER FOR AIR REMOVAL DUCT (FT.)
      PITCH= PD * OD / 12.
C PITCH= TUBE PITCH (FT.)
      RSPACE= PITCH * .8660254
C RSPACE= RADIAL TUBE ROW SPACING FROM TUBE CENTERLINE FOR 60 DEG.
C         PITCH(FT.)
      RAD(1)= VOIDID / 2.
C RAD= RADIUS OF DESIGNATED TUBE ROW
      NTUB(1)= PIE * VOIDID / PITCH
C NTUB= NUMBER OF TUBES PER DESIGNATED TUBE ROW
      SUMTUB= NTUB(1)
      NROW= 1
C
      DO 10 I= 2, 250
      ITOT= I
      RAD(I)= RAD(1) + NROW * RSPACE
      NTUB(I)= 2 * PIE * RAD(I) / PITCH
      SUMTUB= SUMTUB + NTUB(I)
      REMAIN= TNT - SUMTUB
      IF(REMAIN) 200,300,100
 100  NROW= NROW + 1
  10  CONTINUE
 200  PART= TNT - SUMTUB + NTUB(ITOT)
      NTUB(ITOT)= PART
 300  TOTRAD= RAD(ITOT)
C
      NROW= ITOT
      DO 20 J=1, ITOT
      RADIUS(J)= RAD(NROW)
```

```
      NTUBES(J)= NTUB(NROW)
      NROW= NROW - 1
 20   CONTINUE
C
      RETURN
      END
C
      SUBROUTINE NONCOND(ODTLM,DN,OD,TNT,NPASS,TOTNC)
C SUBROUTINE NONCOND: (1) CALCULATES THE PRESSURE DISTRIBUTION AROUND
C THE TUBE BUNDLE PERIPHERY, (2) CALCULATES THE MEAN TOTAL PRESSURE
C AROUND THE TUBE BUNDLE PERIPHERY, AND (3) CALCULATES TUBE LENGTH IN
C THE PRESENCE OF NON-CONDENSABLE GASES.
      DIMENSION WSTM(250),LTUBE(250)
      REAL LREF,L1,LANE,LTUBE,MIXVISC,NTUBES,LSEC
      COMMON STMLD,TSAT,PSAT,STM,TOTRAD,ITOT,LANE,QSTM,PD,EFFL,
     1       TOTL,QT,AVGU,RADIUS(250),NTUBES(250)
      DATA PIE/3.1415927/
      HFG(PCOND)= 1069.185 - 26.72568 * PCOND + 6.659357 * PCOND**2 -
     1            .9288262 * PCOND**3 + .05023536 * PCOND**4
      DENSTM(PCOND)= 1 / (1346.986 - 982.9167 * PCOND + 326.8442 *
     1       PCOND**2 - 49.5767 * PCOND**3 + 2.781686 * PCOND**4)
      TCOND(PCOND)= 46.66471 + 39.7291 * PCOND - 7.869679 * PCOND**2
     1       + .8627231 * PCOND**3 - .03663604 * PCOND**4
      STMVISC(PCOND)= 4.87475E-5 * TCOND(PCOND) + .02191
      AIRVISC(PCOND)= 6.0E-5 * TCOND(PCOND) + .04
      RE(PCOND)= DENSTM(PCOND) * DE * V * 3600 / STMVISC(PCOND)
      F(PCOND)= .0436 / RE(PCOND)**.186
C
C FIRST THE STAGNATION OR TOTAL PRESSURE AT ZONE 1 IS CALCULATED. ALL
C PRESSURE LOSSES AROUND THE TUBE BUNDLE PERIPHERY ARE BASED ON THE
C PRESSURE IN ZONE 1.
C
      PCOND= PSAT
      PT= PSAT + .00021977 * DENSTM(PCOND) / QSTM * STM**2
C PT=     TOTAL PRESSURE (IN. HG) ABSOLUTE
C PSAT=   SPECIFIED CONDENSER PRESSURE (STATIC,IN. HG) ABSOLUTE
```

```
C STM=    MAXIMUM STEAM LANE ENTRANCE VELOCITY (FT/SEC)
C DENSTM= STEAM DENSITY (LB/FT**3)
        DPEN= 1.09885E-5 * DENSTM(PCOND) / QSTM * STM**2
C DPEN= ENTRANCE PRESSURE LOSS (IN.HG)
        DE= 2 * TOTL * LANE / (LANE + TOTL)
C DE= EQUIVALENT DIAMETER OF FLOW AREA(THE RATIO OF FLOW AREA TO
C     WETTED PERIMETER MULTIPLIED BY 4) (FT.)
C
        PL1= PT - DPEN
C PL1= PRESSURE OF CONDENSER ALONG SECTOR NR.1 (IN.HG) ABSOLUTE
        PCOND= PL1
        LREF= TOTRAD * PIE / 6.
        L1=   LREF * 2.
C
        DO 10 J= 1, 3
        CONST1= 9.2593E-5 * QT / (TOTL * LANE)
        V2=     CONST1 / (HFG(PCOND) * DENSTM(PCOND))
C V2= STEAM LANE VELOCITY AT CONDENSER SECTOR NR.- 2
        V= V2
        DP1= 8.7908E-4 * DENSTM(PCOND) / QSTM * F(PCOND)*L1*V2**2/DE
        PCOND= PL1 - DP1
        PL2=   PCOND
  10    CONTINUE
C
        DO 20 J= 1, 3
        V3= .5 * CONST1 / (HFG(PCOND) * DENSTM(PCOND))
        V=  V3
        DP2= 8.7908E-4 * DENSTM(PCOND) / QSTM * F(PCOND)*L1*V3**2/DE
        PCOND= PL2 - DP2
        PL3=   PCOND
  20    CONTINUE
        DO 30 J= 1, 3
        V4= .25 * CONST1 / (HFG(PCOND) * DENSTM(PCOND))
        V=  V4
        DP3= 8.7908E-4 * DENSTM(PCOND) / QSTM*F(PCOND)*LREF*V4**2/DE
        PCOND= PL3 - DP3
```

```
        PL4=   PCOND
  30    CONTINUE
        PDM= (LREF * (PL1 + PL4) + L1 * (PL2 + PL3)) / (3 * L1)
        PCOND= PDM
        TPDM= TCOND(PCOND)
C
C * * * CALCULATION OF TUBE LENGTH IN THE PRESENCE OF NON- CONDENSABLE
C                              GASES. * * *
        FAIR= 10. + .1823E-3 * STMLD - .975E-10 * STMLD**2
C FAIR= AIR FLOW (LB/HR)
        WAWV= FAIR / STMLD
C WAWV= RATIO OF WEIGHT OF AIR-TO-WEIGHT OF STEAM AT CONDENSER INLET
C
        FLUX=  QT / (TNT * PIE * DN * EFFL)
        QTUBE= QT / TNT
        TFLUX= FLUX / TNT
C QTUBE= AVERAGE TUBE HEAT LOAD (BTU/HR)
C FLUX=  CONDENSER HEAT FLUX (BTU/HR-FT**2)
C TFLUX= AVERAGE TUBE HEAT FLUX (BTU/HR-FT**2)
C TNT=   TOTAL NUMBER OF TUBES
C
        PCOND=  PSAT
        REMAIN= QT / (QSTM * HFG(PCOND))
        WAIR=   REMAIN * WAWV
C
        DO 75 NROW= 1, ITOT
        QROW= QTUBE * NTUBES(NROW)
        WSTM(NROW)= REMAIN
C WSTM= STEAM FLOW ENTERING SPECIFIED TUBE ROW (LB/HR)
C WAIR= AIR FLOW ENTERING SPECIFIED TUBE ROW (LB/HR)
        PCOND= PDM
        PROW= PCOND
        DP= 9.427E-6 / (PCOND * 70.73) * TCOND(PCOND)**.8
C DP= COEFFICIENT OF VAPOR DIFFUSION OF STEAM IN AIR
        EW= WAIR / (WAIR + WSTM(NROW))
C EW= WEIGHT RATIO OF AIR-TO-MIXTURE
```

```
      PMPV= 1. + 0.622 * EW / (1. - EW)
C PMPV= PARTIAL PRESSURE MIXTURE-TO-PARTIAL PRESSURE VAPOR
      EV= 1. - 1. / PMPV
C EV= PA/PM: PARTIAL PRESSURE OF AIR IN BULK OF MIXTURE
      MIXVISC= ((1-EV) * STMVISC(PCOND) + 1.61*EV*AIRVISC(PCOND))
     1           / (1. + .61 * EV)
C MIXVISC= ABSOLUTE VISCOSITY OF MIXTURE (LBM/HR-FT)
      PERIM= RADIUS(NROW) * PIE * 2
      A= (PERIM - NTUBES(NROW) * OD / 12.) * TOTL
C A= FLOW AREA FOR SPECIFIED TUBE ROW (FT**2)
      REM= (WSTM(NROW) + WAIR) * OD / (12. * A * MIXVISC)
C REM= REYNOLDS NUMBEROF VAPOR-GAS MIXTURE
      IF(REM .GT. 350) THEN
      AFACT= .82
      ELSE
      AFACT= .52
      END IF
      BFACT= 18745.3  * DP*AFACT*REM**.5 *(PCOND * 70.73)**.3333
     1  * HFG(PCOND)**1.66667 / (OD * EV**.6) * (DENSTM(PCOND) /
     2  ((459.67 + TCOND(PCOND)) * QSTM))**.66667
      FLX= TFLUX * NTUBES(NROW)
      QFLUX= FLX
C
      DO 80 J= 1, 5
      DTNEW= TCOND(PCOND) - TSAT + ODTLM
      QFLUX= DTNEW / (QFLUX**.5 / BFACT**1.5 + 1 / AVGU)
   80 CONTINUE
      UNEW= 1 / (QFLUX**.5 / BFACT**1.5 + 1 / AVGU)
      LTUBE(NROW)= QROW * 12. / (PIE * DN * NTUBES(NROW) * UNEW
     1                * DTNEW)
      PCOND= PSAT
      REMAIN= (QT - QROW) / (QSTM * HFG(PCOND))
C
      IF(NROW .EQ. 1) THEN
      QZ= QT
      ELSE
```

```
      QT= QT - QROW
      END IF
C
      IF(NROW .EQ. 1) THEN
      PRESS= PDM
      ELSE
      END IF
C
      PCOND= PRESS
      VROW=  REMAIN / (DENSTM(PCOND) / QSTM * A * 3600.)
      RET=   150. * DENSTM(PCOND)/QSTM * OD * VROW/STMVISC(PCOND)
C
      IF(RET .GT. 1000.) THEN
      FT= .591 * RET**-.136 * PD**-1.137
      ELSE
      FT= 65.5 * RET**-.702 * PD**-4.257
      END IF
C
      DPT=   7.32567E-5 * FT * DENSTM(PCOND) / QSTM * VROW**2
      PRESS= PRESS - DPT
   75 CONTINUE
      SUML= 0.0
C
      DO 90 J= 1, ITOT
      LSEC= LTUBE(J) * NTUBES(J)
      SUML= SUML + LSEC
   90 CONTINUE
C
      TOTNC= SUML / TNT
      QT= QZ
      RETURN
      END
```

# APPENDIX B

## CONTENTS


Page

Properties of Multiple-Helix Internal Ridged Tubes. . . 107

Friction Factor Characteristics of Multiple-Helix Internal Ridged Tubes . . . . . 108

Friction Factor Curves for Multiple-Helix Internal Ridged Tubes . . . . . 108

Operands for Friction Factor Equation versus Geometric Aspect Ratio e/1 for Multiple-Helix Internal Ridging

$p/d_i \geq 0.36$. . . . . 109

$p/d_i < 0.36$. . . . . 109

Waterbox and Tube Endlosses Single Pass Condensers . . 110

Waterbox and Tube End Losses Double Pass Condensers. . 111

TABLE 4- Properties of Multiple-Helix Internal Ridged Tubes

| Symbol | Tube no. | Nominal diameter, in | Root diameter $d_r$, in | Fin count, fins/in | Outside area, $ft^2/ft$ | ID (max) $d_i$, in | No. of starts |
|---|---|---|---|---|---|---|---|
| △ | 12 | 3/4 | 0.623 | (Stripped) | 0.163 | 0.573 | 5 |
| ▷ | 30 | 3/4 | 0.627 | 26.4 | 0.640 | 0.575 | 6 |
| ▽ | 22 | 3/4 | 0.625 | 26 | 0.640 | 0.569 | 6 |
| ◁ | 27 | 3/4 | 0.622 | 26.5 | 0.640 | 0.572 | 6 |
| ▲ | 28 | 3/4 | 0.624 | 26 | 0.640 | 0.573 | 5 |
| ▶ | 31 | 3/4 | 0.624 | 26.6 | 0.640 | 0.576 | 6 |
| ▼ | 29 | 3/4 | 0.625 | 26.1 | 0.640 | 0.575 | 5 |
| ◀ | 38 | 3/4 | 0.633 | 38.0 | 0.830 | 0.572 | 8 |
| ◇ | 37 | 3/4 | 0.624 | 38.5 | 0.901 | 0.574 | 10 |
| | 40 | 3/4 | 0.627 | 27.3 | 0.689 | 0.561 | 10 |
| ⬡ | 41 | 3/4 | 0.628 | 38.1 | 0.852 | 0.572 | 12 |
| ◁ | 9 | 3/4 | 0.628 | 38.5 | 0.901 | 0.575 | 10 |
| □ | 21 | 7/8 | 0.740 | 26 | 0.640 | 0.684 | 6 |
| ■ | 19 | 7/8 | 0.745 | (Stripped) | 0.195 | 0.692 | 6 |
| | 44 | 3/4 | 0.627 | 41.0 | | 0.574 | 6 |
| | 43 | 3/4 | 0.628 | 41.0 | | 0.573 | 10 |
| ○ | 42 | 3/4 | 0.626 | 41.0 | 0.901 | 0.573 | 10 |
| | 46 | 3/4 | 0.627 | 41.2 | | 0.573 | 10 |
| | 45 | 3/4 | 0.627 | 41.2 | | 0.577 | 10 |
| ○ | 13 | 1 | 0.883 | 26 | 0.841 | 0.820 | 6 |
| ◒ | 32 | 1 | 0.877 | 26 | 0.871 | 0.825 | 6 |
| ◒ | 25 | 1 | 0.880 | 26 | 0.841 | 0.816 | 6 |
| ◒ | 24 | 1 | 0.878 | 26 | 0.841 | 0.814 | 6 |
| ◒ | 23 | 1 | 0.880 | 26 | 0.841 | 0.816 | 6 |
| ◐ | 26 | 1 | 0.864 | 27.3 | 0.841 | 0.815 | 6 |

| Symbol | Internal ridging: Height $e$, in | Internal ridging: Pitch $p$, in | Internal aspect ratios: $e/d_i$ | Internal aspect ratios: $e/p$ | Internal aspect ratios: $p/d_i$ | ID of envelope tube, in |
|---|---|---|---|---|---|---|
| △ | 0.0125 | 0.475 | 0.0218 | 0.0263 | 0.829 | 1.000 |
| ▷ | 0.0162 | 0.279 | 0.0282 | 0.0581 | 0.485 | 1.000 |
| ▽ | 0.0165 | 0.320 | 0.0288 | 0.0516 | 0.558 | 1.000 |
| ◁ | 0.017 | 0.385 | 0.0297 | 0.0442 | 0.673 | 1.000 |
| ▲ | 0.019 | 0.475 | 0.0332 | 0.0400 | 0.829 | 1.000 |
| ▶ | 0.0198 | 0.287 | 0.0344 | 0.0690 | 0.498 | 1.000 |
| ▼ | 0.0207 | 0.469 | 0.0360 | 0.0441 | 0.816 | 1.000 |
| ◀ | 0.018 | 0.212 | 0.0315 | 0.0849 | 0.371 | 1.000 |
| ◇ | 0.0200 | 0.166 | 0.0348 | 0.1205 | 0.289 | 1.000 |
| | 0.0175 | 0.170 | 0.0312 | 0.1029 | 0.303 | 1.000 |
| ⬡ | 0.0215 | 0.138 | 0.0376 | 0.1558 | 0.241 | 1.000 |
| ◁ | 0.0204 | 0.191 | 0.0355 | 0.1070 | 0.332 | 1.000 |
| □ | 0.017 | 0.391 | 0.0249 | 0.0435 | 0.572 | 1.2348 |
| ■ | 0.017 | 0.285 | 0.0246 | 0.0596 | 0.412 | 1.000 |
| | 0.021 | 0.207 | 0.0366 | 0.101 | 0.361 | — |
| | 0.021 | 0.124 | 0.0366 | 0.169 | 0.216 | — |
| ○ | 0.024 | 0.0949 | 0.0419 | 0.253 | 0.166 | 1.000 |
| | 0.021 | 0.094 | 0.0367 | 0.223 | 0.164 | — |
| | 0.015 | 0.094 | 0.0260 | 0.1596 | 0.163 | — |
| ○ | 0.0178 | 0.333 | 0.0217 | 0.0535 | 0.406 | 1.5936 |
| ◒ | 0.0193 | 0.335 | 0.0234 | 0.0576 | 0.406 | 1.5936 |
| ◒ | 0.0205 | 0.340 | 0.0253 | 0.0603 | 0.420 | 1.5936 |
| ◒ | 0.0205 | 0.330 | 0.0252 | 0.0621 | 0.405 | 1.5936 |
| ◒ | 0.0205 | 0.338 | 0.0251 | 0.0607 | 0.414 | 1.5936 |
| ◐ | 0.021 | 0.340 | 0.0258 | 0.0618 | 0.417 | 1.5936 |

TABLE 5 Friction Factor Characteristics of Multiple-Helix Internal Ridged Tubes

| Tube no. | $m$ | $r$ |
|---|---|---|
| 12 | 0.762 | 0 |
| 30 | 0.64 | −0.00039 |
| 22 | 0.697 | −0.00017 |
| 27 | 0.72 | −0.00026 |
| 28 | 0.72 | −0.00028 |
| 31 | 0.61 | −0.00095 |
| 29 | 0.68 | −0.00064 |
| 38 | 0.61 | −0.00109 |
| 37 | 0.54 | −0.00392 |
| 40 | 0.627 | −0.00080 |
| 41 | 0.57 | −0.00259 |
| 9 | 0.59 | −0.00197 |
| 21 | 0.70 | −0.00014 |
| 19 | 0.626 | 0 |
| 44 | 0.53 | −0.00180 |
| 43 | 0.55 | +0.00017 |
| 42 | 0.58 | +0.00750 |
| 46 | 0.54 | +0.00275 |
| 45 | 0.52 | −0.00159 |
| 13 | 0.63 | +0.00024 |
| 32 | 0.70 | +0.00090 |
| 25 | 0.645 | +0.00032 |
| 24 | 0.64 | −0.00035 |
| 23 | 0.637 | +0.00028 |
| 26 | 0.64 | +0.00018 |

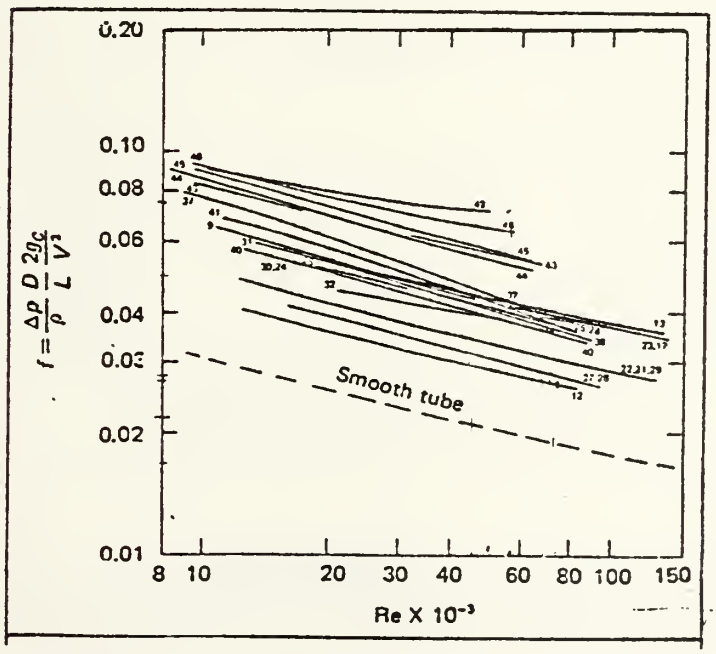



FIGURE 17 Friction Factor Curves for Multiple-Helix Internal Ridged Tubes

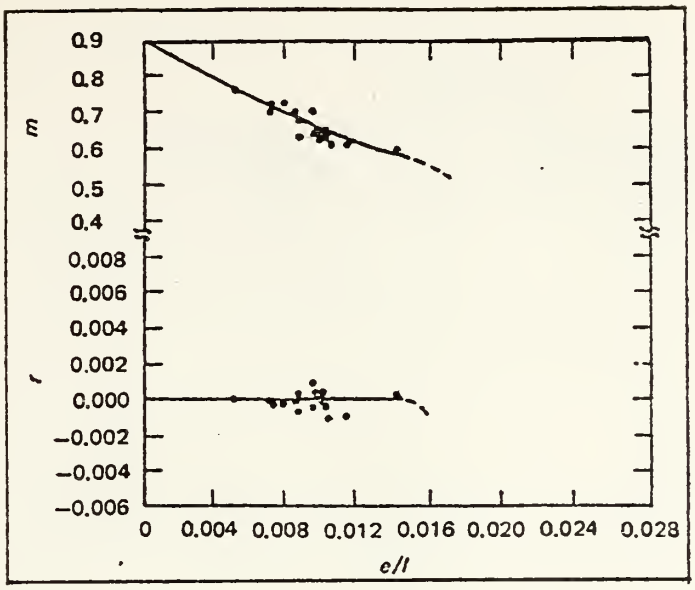



FIGURE 18a
Operands for friction factor equation versus geometric aspect ratio e/1 for multiple-helix internal ridging with $p/d_i \geq 0.36$

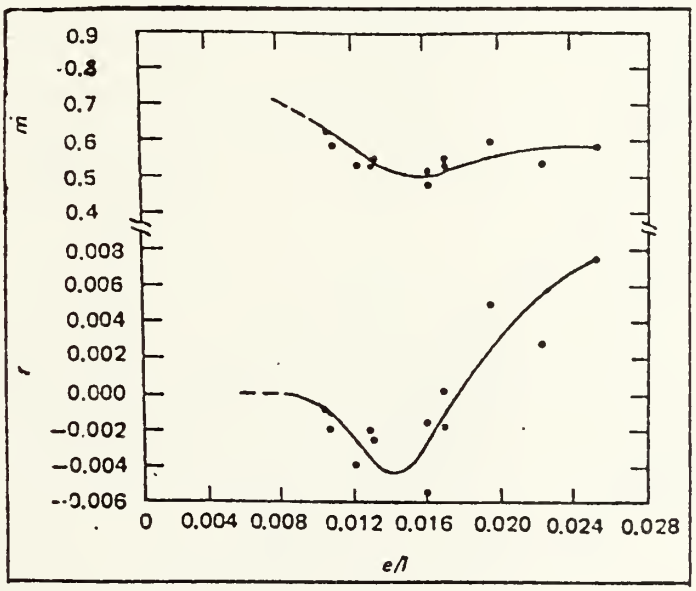



FIGURE 18b
Operand for friction factor equation versus geometric aspect ratio e/1 for multiple-helix internal ridging with $p/d_i < 0.36$

FIGURE-19

## WATERBOX AND TUBE END LOSSES SINGLE PASS CONDENSERS

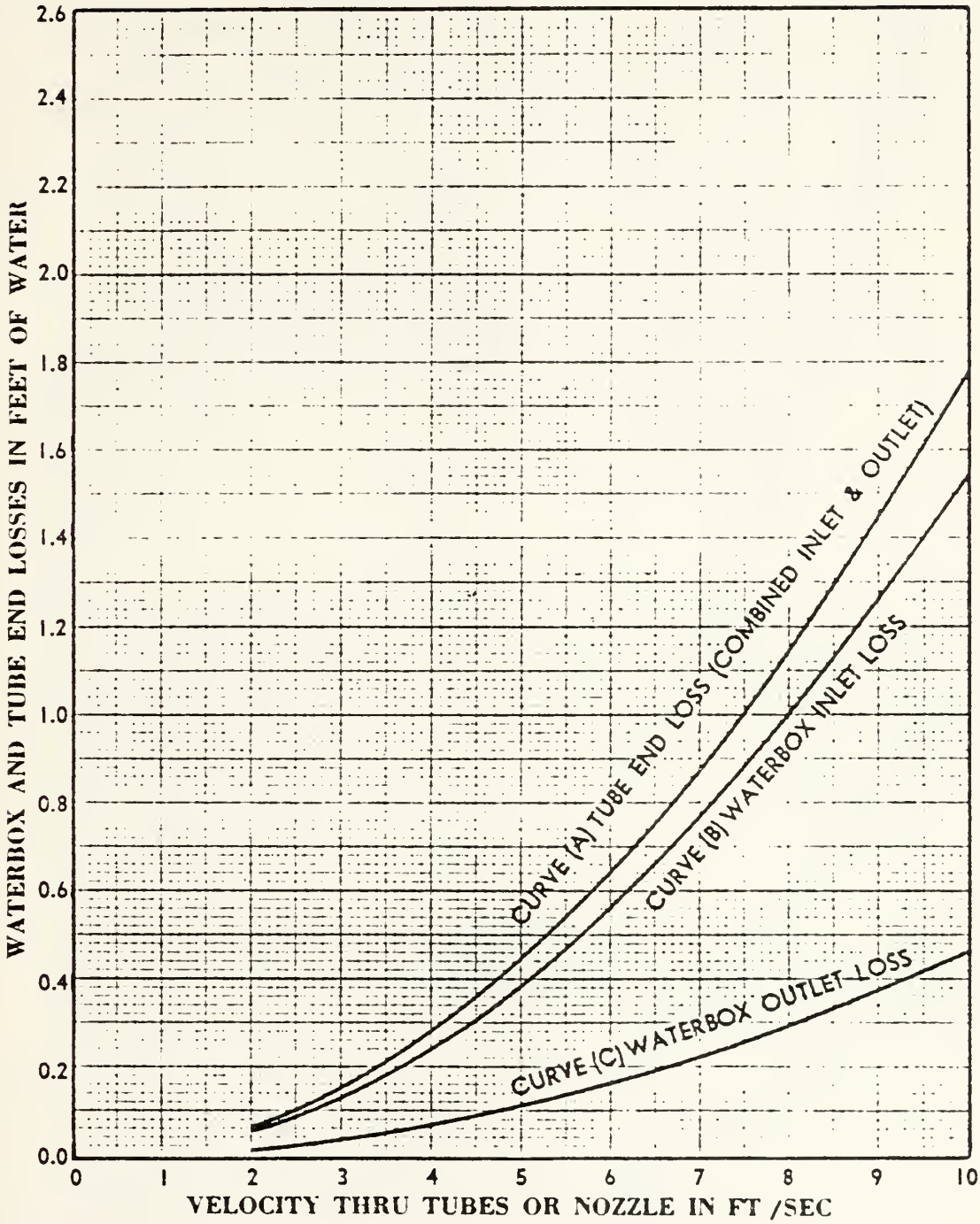
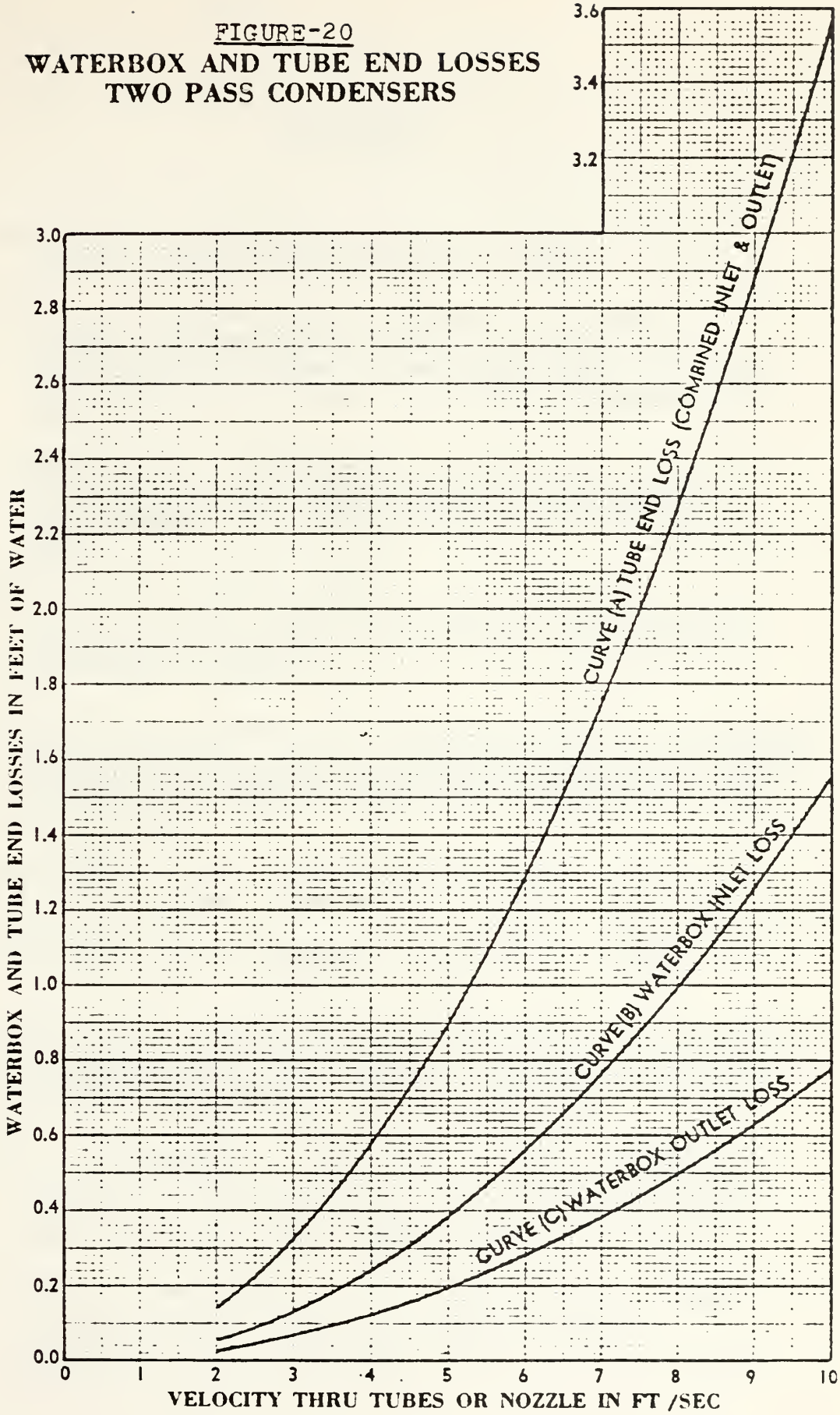


FIGURE-20

## WATERBOX AND TUBE END LOSSES TWO PASS CONDENSERS



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